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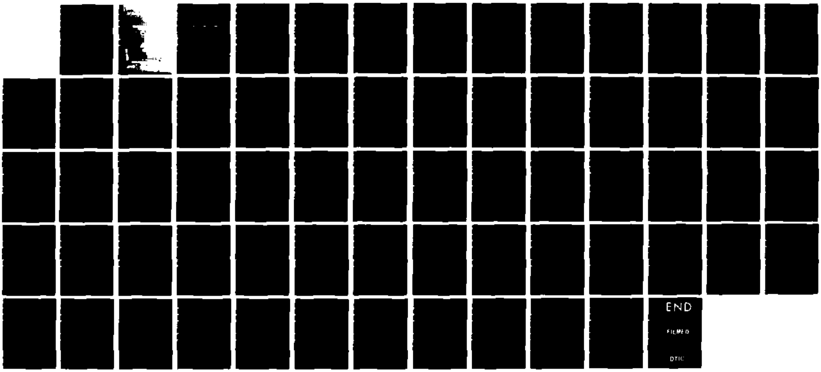

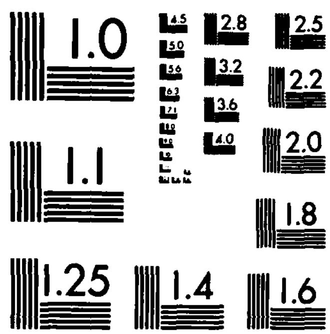

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ANALYSIS OF ARMOURED PERSONNEL CARRIER HEAT LOSS

by

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DEFENCE RESEARCH ESTABLISHMENT OTTAWA

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ABSTRACT

This report describes the heating properties of the Canadian Forces Armoured Personnel Carrier (APC). In the theoretical analysis, the APC is modelled as a box, having various insulating covers on the surfaces. The rate of heat transfer from the APC is then estimated for each insulation configuration using empirical heat transfer coefficients and measured thermal conductances. All assumptions, equations and physical data used in the analysis are presented as is a listing of the FORTRAN computer program which was used to solve the heat transfer equations. Thus, the reader may use this report as a guide to performing the heat transfer analyses for similar problems. Field measurements of internal air temperature and of heat flow-rates were made using two APC's in various insulating configurations. The measured heat loss from an uninsulated APC was in close agreement to the heat loss predicted by the theoretical analysis. However, the measured heat loss from an APC insulated with a covering developed by the Defence Research Establishment Suffield was found to be significantly greater than that predicted by the theoretical analysis. Theories are advanced on possible sources of error and discrepancy between the model predictions and the measured values of temperature and heat flow. Heat loss due to ventilation is also examined briefly.

RÉSUMÉ

Il s'agit d'un rapport décrivant les propriétés thermiques du véhicule blindé de transport de troupes (VBTP) des Forces canadiennes. Dans l'analyse théorique, le VBTP est représenté par une boîte recouverte de diverses enveloppes isolantes. On estime ensuite le taux de transfert de la chaleur qui se dégage du véhicule dans chaque cas, utilisant à cet effet des coefficients empiriques de transfert de la chaleur et les conductances thermiques mesurées. Toutes les hypothèses, les équations et les données sur les propriétés physiques utilisées aux fins de l'analyse ainsi que le programme d'ordinateur en langage FORTRAN ayant servi à résoudre les équations de transfert thermique y sont présentés. Ainsi, ce rapport peut servir de guide au lecteur pour effectuer des analyses de transfert thermique applicables à des problèmes similaires. La température de l'air à l'intérieur de deux VBTP recouverts de diverses enveloppes isolantes ainsi que le flux de chaleur qui se dégageait des véhicules ont été mesurés en situation réelle. La perte de chaleur du VBTP non isolé tel que mesurée concorde très bien avec les prévisions théoriques. Par contre, il y ent un écart considérable entre les résultats expérimentaux et le modèle théorique dans le cas du VBTP avec l'enveloppe isolante du centre de recherches pour la défense, Suffield. Des théories sont avancées sur les causes possibles d'erreur et les divergences qui existent entre les prédictions théoriques et les mesures obtenues. La perte de chaleur attribuable à la ventilation est également examinée en bref.

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LIST OF VARIABLES

A, A_i - surface area.

$[A]$ - a matrix whose elements are composed of the various thermal resistances of the enclosure.

$A(i,j)$ - elements of the matrix $[A]$.

$[B]$ - a vector whose elements are composed of the known air temperatures and the thermal resistances of the enclosure.

$B(i)$ - an element of the vector $[B]$.

c - an empirical heat transfer constant.

C_p - fluid heat capacity at constant pressure.

F_{ij} - the viewfactor from surface i to surface j.

g - the acceleration due to gravity.

Gr - the Grashof Number, $g\beta L^3(T_i - T_\infty)/\nu^2$.

h - the convective and conductive heat transfer coefficient.

i,j - integer variables designating surfaces.

k - fluid thermal conductivity.

L - the characteristic length of a surface.

n - an empirical heat transfer constant.

Nu - the Nusselt Number, hL/k.

Pr - the Prandtl Number, $C_p \nu/k$.

q - the fluid volumetric flowrate.

Q - the total rate of heat loss from an enclosure from all sources.

Q_i - the net radiant heat transfer rate leaving surface i.

Q_v - the rate of heat loss from an enclosure due to air exchange with the environment.

r_{ij} - the radiative thermal resistance between surface i and surface j.

Re_x - the local Reynold's Number based on position, Ux/ν.

Re_L - the Reynold's Number based on the characteristic length, UL/ν.

R_{eq} - the equivalent thermal resistance of the enclosure.

R_{ij} - the jth convective or conductive thermal resistance of surface i.

R'_{ij} - the net jth thermal resistance of surface i due to the action of convective and radiative thermal resistance in parallel.

LIST OF VARIABLES (Cont'd)

St - Stanton Number, $h/(\rho C_p U)$.
t - thickness.
t_i - measured local internal air temperature.
[T] - a vector whose elements are the desired internal wall surface temperatures.
T_a - the volume weighted internal air temperature.
T_i - an element of the vector [T].
T_m - a mean temperature.
T_o - the internal vehicle air temperature.
T_∞ - the external ambient air temperature.
U - the free-stream fluid velocity over a surface.
V - the total volume of an enclosure.
V_i - a local volume element associated with the temperature t_i.
x - the length from the leading edge of a flat plate to the point of interest.
β - fluid coefficient of thermal expansion.
ε - the emissivity of a surface.
ν - the fluid kinematic viscosity.
ρ - the fluid density.
σ - the Boltzman Constant, 5.669×10^{-8} [W/m^2K^4].

1.0 INTRODUCTION

The Canadian Forces (CF) have a commitment to operate in regions in which the weather can be as much a problem as any human adversary. The CF are frequently on exercise in Canada's High Arctic and in northern Norway where temperatures can easily reach -40°C. Problems associated with the environment can often be overcome by making use of specialized equipment and relevant training.

Mechanized infantry units of the CF are using Armoured Personnel Carriers (APC's) to increase their mobility and effectiveness. Unfortunately, under the severe cold weather conditions which can occur in the Arctic, vehicle problems are abundant. In an attempt to reduce these vehicle problems, conserve fuel and keep the occupants of the APC's warm, Defence Research Establishment Suffield (DRES) was tasked to devise an insulating cover for the APC. The primary goal of the task is to produce a cover which will maintain the internal vehicle temperature at a level that will facilitate starting of the vehicle in extremely cold weather, with a minimum of auxiliary heating.

DRES approached the Environmental Protection Section (EPS) of Defence Research Establishment Ottawa (DREO) for advice on the thermal properties for a protective covering. The EPS is devoted to research in the field of protecting the man from various aspects of the environment, but specializing in cold weather protection. Protecting a man from the cold has many similarities to the problem at hand.

This report presents an idealized analysis of an APC which provides the heat transfer information which DRES requires to choose its most effective course of action in developing a vehicle covering. This report gives the reasoning and procedures behind each step or assumption of the analysis. Thus, the reader may use this report as a guide for performing similar analyses of heat transfer from enclosures. With a minimal amount of work, the interested reader can perform the analysis under different sets of ambient conditions, with different insulation values or under different sets of assumptions as required by particular problems. The algorithm used to perform the actual calculation of the heat transfer and temperatures (Appendix C) is written in standard FORTRAN-77 and was used on a Honeywell computer under a CP-6 operating system. For the layman, only a basic understanding of FORTRAN and computers is required in order to apply the heat transfer analysis to similar applications. The equations used to determine the heat transfer coefficients are given through the text in their appropriate sections.

Experimental measurements of the effectiveness of the DRES prototype covering, were made at DREO and are included in this report. The purpose of these measurements is mainly to validate the theoretical estimate of the predicted effectiveness of an insulated cover. A secondary function of the experimental measurements is to obtain a quantitative evaluation of the DRES prototype covering.

2.0 HEAT TRANSFER ANALYSES

2.1 THEORETICAL ANALYSIS

Heat transfer in the vehicle and from the vehicle can occur in four ways:

(1) Convection from a surface to the surrounding fluids;

(2) Radiation from a surface directly to another surface;

(3) Conduction through a body from one side to another;

(4) Convection by air exchange between the vehicle and the ambient air (ventilation).

The following analysis uses the first three heat transfer mechanisms to predict heat flow rates and the effects of adding insulation to the surfaces of the vehicle. As ventilation is essentially independent of the vehicle insulation, its analysis will be covered in a separate section. The ventilatory heat loss acts in parallel with the other three heat loss mechanisms and thus, all four mechanisms must be considered simultaneously when determing the actual effectiveness of an insulating cover.

Dimensions taken from a test vehicle at DREO were used in the idealized model of an APC shown in Figure 1.

Convective heat transfer coefficients depend upon the fluid properties. As these properties are temperature dependent, it is customary to evaluate them at a temperature which is the average of the wall and fluid temperatures. For the purpose of this study, the outside ambient air temperature is assumed to be -40°C while the internal air temperature is assumed to be 15°C. Fortunately, the physical properties of air (Appendix A) are only mildly dependent upon temperature in the above temperature range and even sizable deviations from the assumed temperatures have only

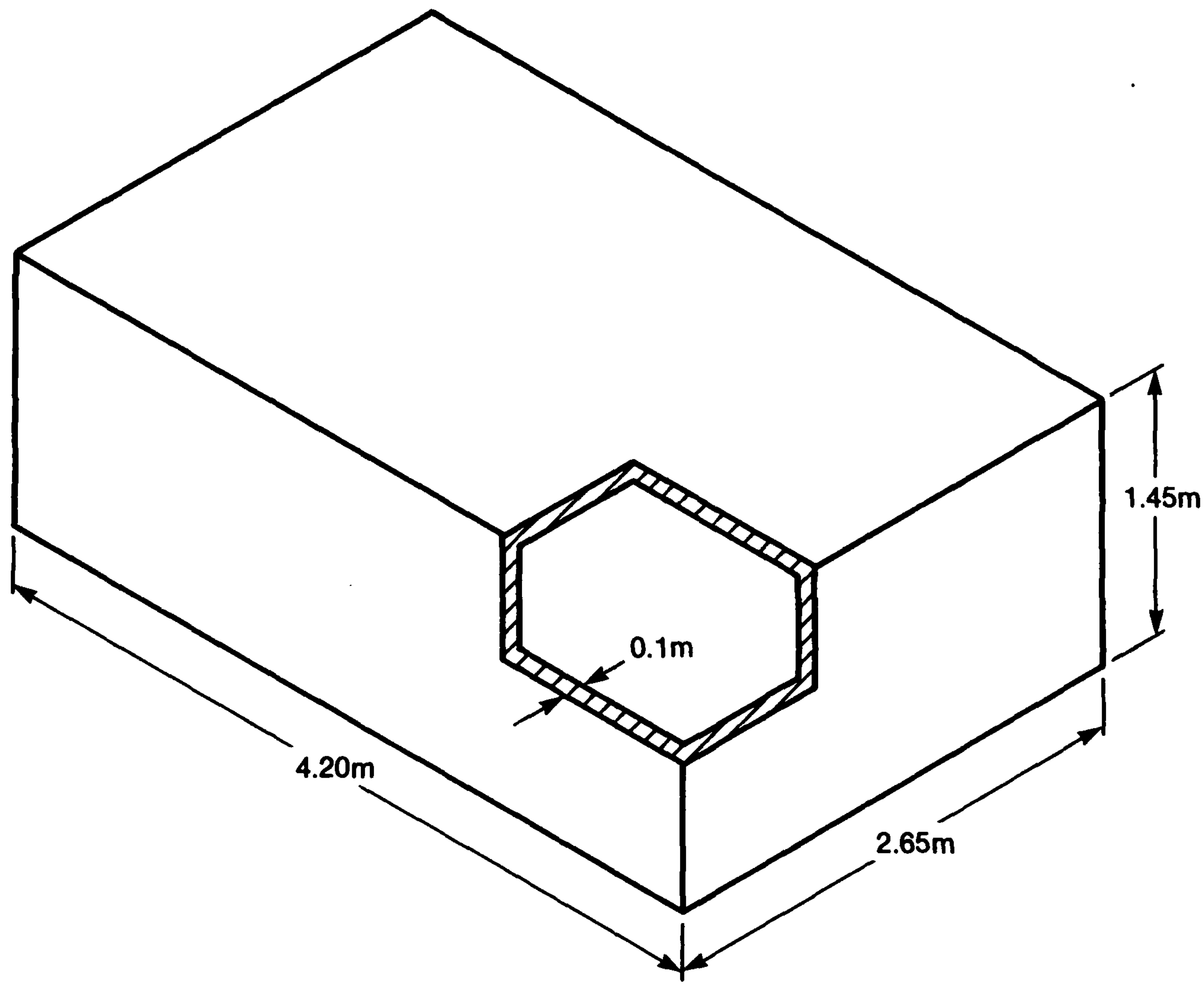

Figure 1: Enclosure Model Used to Analyse the Heat Loss from An Armoured Personnel Carrier.

small effects upon the magnitudes of the heat transfer coefficients. As the internal wall temperatures are unknown, it is initially assumed that all the walls are at -12.5°C. Using this temperature, a heat transfer coefficient may be calculated. If the subsequent evaluation of the predicted wall temperature varies by more than 5°C from this value, the heat transfer coefficients will be re-evaluated at this new wall temperature and the calculation of wall temperatures will be performed again. This iterative procedure may be repeated as often as necessary, re-evaluating the heat transfer coefficients using the latest wall temperature prediction.

Several dimensionless groups are used in characterizing heat transfer and fluid flow in an attempt to generalize empirical results. The dimensionless groups of interest to this study are:

Reynolds No.: $Re = \frac{UL}{\nu}$ Ratio of (Inertial Forces) to (Viscous Forces). (1)

Nusselt No. : $Nu = \frac{hL}{k}$ Ratio of (convective heat transfer rate from the surface to the fluid) to (Conductive heat transfer through the liquid). (2)

Stanton No. : $St = \frac{h}{C_p U}$ Ratio of (Energy transfer rate to the fluid) to (Energy flowrate of the fluid due to the fluid velocity). (3)

Prandtl No. : $Pr = \frac{C_p \nu}{k}$ Ratio of (Molecular diffusivity of momentum) to (Molecular diffusivity of heat). (4)

Grashof No. : $Gr = \frac{g\beta L^3 (T_i - T_\infty)}{\nu^2}$ (5)

It is convenient to analyze the problem by making an electrical analogy where the temperature difference (which is the driving potential) is equivalent to the voltage drop in the circuit; the heat flow rate is equivalent to electrical current; and the thermal heat transfer coefficients and conductances can be formulated as thermal resistances corresponding to electrical resistances in the analogy.

Figure 2 is a graphical representation of the electrical circuit which is equivalent to the thermal problem of a heated enclosure which has several different layers of thermal resistance between the interior and the exterior. Each large "R" represents a conductive thermal resistance; each small "r" represents a radiative thermal resistance; each large "R'" represents an equivalent thermal resistance of convective and radiative resistances which act in parallel.

The T_{ij} refer to surface temperatures: T_{i1} are the surface temperatures of the interior walls; T_{i2} are the surface temperatures of the

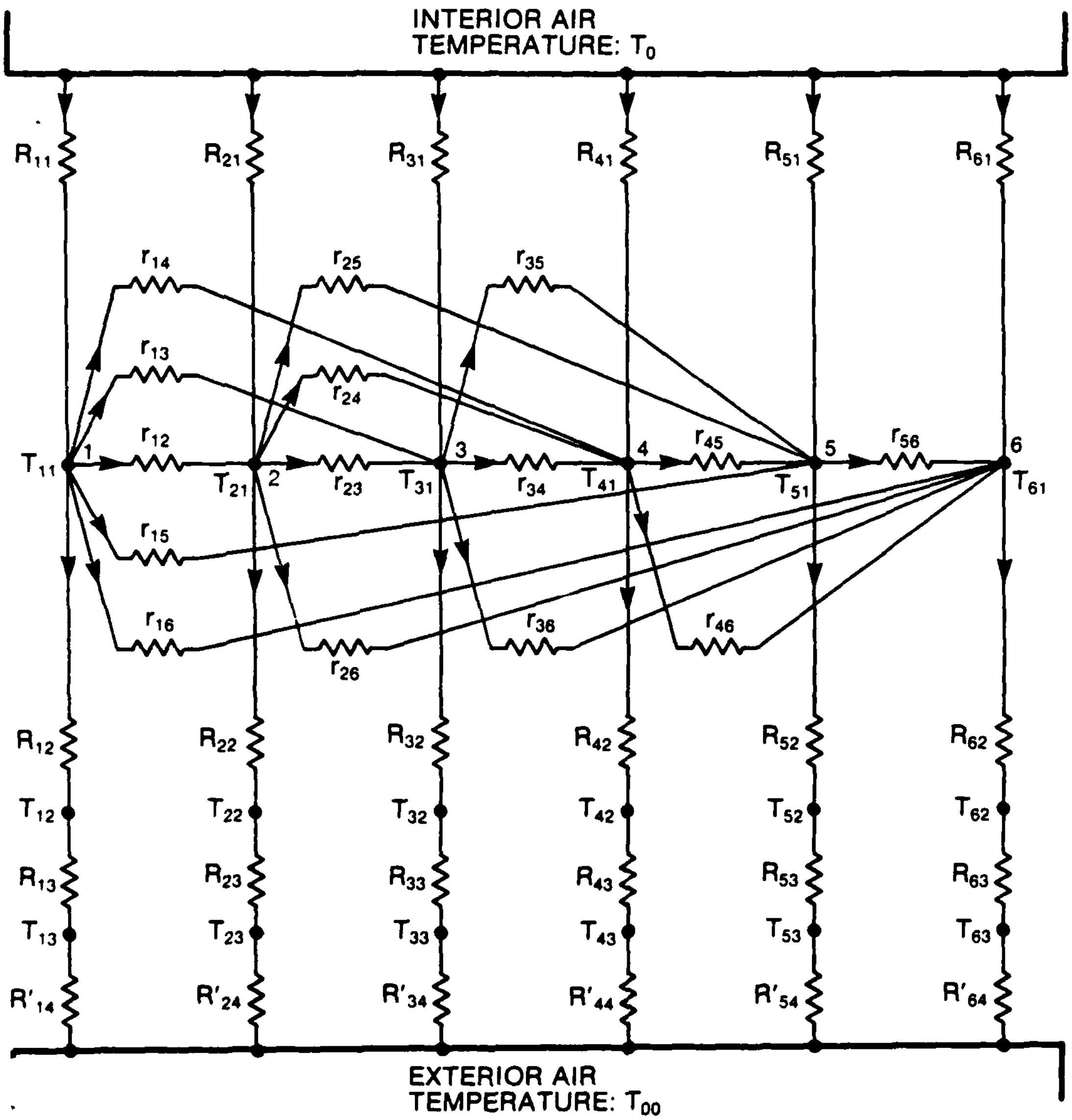

Figure 2: Electrical Circuit Analogy of the Heat Transfer for the Armoured Personnel Carrier.

The electric heaters typically produced 1300 [W] each. Each heater had an electric fan which produced a typical mass flowrate of 7.1×10^{-2} [kg/s]. One of the three heaters was positioned so that the air was blown horizontally while the other two were positioned so that the air was blown upwards. All three electric heaters in each vehicle were placed at the front of the crew compartment adjacent to the vehicle heater.

The thermistors were distributed throughout the vehicle so as to give a representative temperature distribution. The location of each thermistor corresponded as closely as possible to the same location in each vehicle. A typical thermistor distribution is shown in Figure 3. The heat flow discs were placed on the floor in the middle of the cargo space. Both of the engine compartment doors were removed to allow the internal air to circulate into the engine compartment.

Several sets of measurements were taken, each measurement period lasting from 1800 hrs to 0700 hrs the next day. In case 1 of the measurements, neither of the vehicles had additional insulation. This was done to observe any differences in the two vehicles, or in the data collection and interpretation technique. In cases 2a and 2b, the differences between the heat transfer properties of an uninsulated APC and an APC insulated with the DRES "Blanket" are shown. In case 3, air was allowed to infiltrate into the vehicle through the hatches to observe ventilatory heat losses.

The data were analysed by assigning a volume element to each of the thermistors in the vehicle, so that the sum of all the volume elements equalled the total volume of the APC. Then, using these volumes, the volume weighted mean temperature of the interior of the APC was calculated [7] by:

$$T_a = \sum_i t_i V_i / V \tag{23}$$

The equivalent thermal resistance (R_{eq}) on a unit area basis is then calculated by dividing the difference between the mean temperature and the ambient temperature by the rate of heat input:

$$R_{eq} = A(T_a - T_\infty)/Q \tag{24}$$

where the total vehicle surface area, A, is assumed to be 42.125 m^2. Using this parameter, the effectiveness of the insulating layer can be assessed. Table 7 of section 3.2 lists the R_{eq} of the vehicles and several other important temperatures found in the experiments. When ventilation of the

2.2 ANALYSIS OF THE EFFECT OF VENTILATION

If any of the hatches are open, or other openings between the interior and exterior of the vehicle exist, thermal energy may be lost by the escape of warm air to the environment which is then replaced by cold ambient air.

If it is assumed that the rate at which warm air escapes from the vehicle is q [m³/s], then the additional vehicle heating required to maintain the same internal temperature is:

$$Q_v = \rho\, C_p q(T_o - T_\infty) \tag{22}$$

It should be stressed that the quantity Q_v will only ensure that the mean internal air temperature will remain unchanged. Incomplete mixing and poor circulation will undoubtebly result in areas of cold drafts and hot stagnant pockets of air.

The only current practical solution to this heat loss problem is to minimize the air exchange with the environment. The lower limit of this minimization is the rate at which air is used by the occupants of the vehicle and by equipment which use air, such as combustion heaters, which are operating in the vehicle.

2.3 EXPERIMENTAL ANALYSIS

Two APC's were used at DREO for the experimental testing of the DRES "Blanket". Each APC was made "air-tight" either by extensive taping over the holes, or by filling the holes with a foam insulating compound. Missing hatch-covers were replaced with a 6 mil plastic sheet firmly secured about the hatch with tape.

After the APC's were made as air-tight as possible, the DRES Blanket was placed on one of the vehicles. Thermistors, heat-flow discs and electrical heaters were then installed inside the vehicles.

TABLE 4c: Numerical Values of Thermal Resistances for Idealized APC: Conductive Heat Transfer for Various Insulating Configurations.

Region		R_{i3}[K/W]				
		Case 1	Case 2	Case 3	Case 4	Case 5
Roof	i=1	0.0	0.0353	0.0353	0.0706	0.1412
Side Walls,	i=2,3	0.0	0.0694	0.0694	0.1387	0.2774
End Walls,	i=4,5	0.0	0.1115	0.1115	0.2229	0.4458
Floor,	i=6	0.0	0.0	0.0520	0.1040	0.2079

Case Description

Case 1: No insulating cover.
Case 2: Idealized DRES "Blanket", no floor insulation.
Case 3: Idealized DRES "Blanket" plus polyethylene foam, floor insulation.
Case 4: Double the insulation thicknesses of Case 3.
Case 5: Quadruple the insulation thicknesses of Case 3.

TABLE 4a: Initial Numerical Values of Thermal Resistances for an Idealized APC: Internal Convective (R_{i1}), Internal Conductive (R_{i2}) and and Exterior Convective/Radiative (R_{i4}) Heat Transfer.

(T_o = 15°C), T_∞ = -40°C, T_m = -40°C, V = 20 m/s).

Region	Designator i	R_{i1} (K/W)	R_{i2} (K/W)	R'_{i4} (K/W)
Roof	1	3.1×10^{-2}	4.5×10^{-5}	2.2×10^{-4}
Side Walls	2,3	5.0×10^{-2}	8.8×10^{-5}	4.3×10^{-4}
End Walls	4,5	8.1×10^{-2}	14.2×10^{-5}	9.5×10^{-4}
Floor	6	12.4×10^{-2}	4.5×10^{-5}	2.2×10^{-4}

TABLE 4b: Numerical Values of Thermal Resistances for an Idealized APC: Internal, Radiative Heat Transfer. (T_m = 15°C).

To / From	r_{ij} [k/W]					
	j=1	j=2	j=3	j=4	j=5	j=6
Roof, i=1	∞	0.1158	0.1158	0.2001	0.2001	0.0392
Side Wall 1, i=2	0.1158	∞	0.2305	0.3660	0.3660	0.1158
Side Wall 2, i=3	0.1158	0.2305	∞	0.3660	0.3660	0.1158
End Wall 1, i=4	0.2001	0.3660	0.3660	∞	1.0813	0.2001
End Wall 2, i=5	0.2001	0.3660	0.3660	1.0813	∞	0.2001
Floor, i=6	0.0392	0.1158	0.1158	0.2001	0.2001	∞

2.1.6 Numerical Evaluation of Individual Thermal Resistances

Tables 4a, 4b and 4c list the numerical values of the thermal resistances of the model, using the heat transfer coefficients calculated in Sections 2.1.1 to 2.1.3 with the appropriate thermal resistance formulae and the radiant thermal resistances of Sections 2.1.4 and 2.1.5. At the exterior surface, the convective and radiant heat transfer act in parallel (assumption 2 of Section 2.1.5). Thus, the equivalent thermal resistance of this pair may be calculated as:

$$R_i' = \frac{1}{\left(\frac{1}{R_i} + \frac{1}{r_i}\right)} = \frac{1}{(A_i \sigma T_m^3 + h_i A_i)} \qquad (21)$$

To study the effects of adding insulation (varying R_{i3} of Figure 2), five different insulating sample cases were studied.

Case 1: No additional insulation (R_{i3} = 0 [K/W], i=1,6);

Case 2: The equivalent of the DRES "Blanket" ($R_{i3} = 0.424/A_i \frac{K}{W}$, $R_{63} = 0 \frac{K}{W}$);

Case 3: A floor insulation equivalent to 2 cm of closed cell polyethylene foam ($R_{63} = 0.441 \frac{m^2K}{W}$) was assumed in addition to the insulation values of Case 2;

Case 4: Double the insulating values of Case 3;

Case 5: Quadruple the insulating values of Case 3.

that all of its interior walls are at similar temperatures, and these temperatures must not be greatly different from the surface temperature of the bodies inside the vehicle. (Note that the surface temperature of a clothed soldier may be significantly different from skin temperature.)

2.1.5 Radiant Heat Transfer - External

In general, the ground and sky temperatures can be significantly different from the ambient air temperature at ground level [6]. The following assumptions were made to simplify the analysis:

1. The emissivity of the surroundings is 1.0;
2. The temperature of the surroundings is the same as the ambient temperature.

These assumptions may be refined if additional information is available. If the analysis is to be applied to warm weather conditions with a significant amount of solar heating, assumption 2 may be invalid and a more detailed analysis may be required.

From the geometry of the model, the viewfactor from each surface of the model to the surroundings is 1. Thus, the radiant heat transfer from surface i to the surroundings will be approximately:

$$Q_i = 4A_i \sigma T_m^3 (T_i - T_\infty) \tag{19}$$

and the thermal resistance for the surface i will be:

$$r_{i\infty} = \frac{1}{4A_i \sigma T m^3} \tag{20}$$

In this analysis, the radiant heat transfer was made linear in temperature in the same manner as in Section 2.1.4. As a first order approximation, T_m is assumed to have the same value as the ambient temperature (-40°C).

Calculated viewfactors (F_{ij}) are given in Table 3. When these values are used with equation (17) to determine the radiant thermal resistance, it is assumed that the emissivity of the surfaces are all unity.

TABLE 3: Viewfactors for Radiative Heat Transfer Between Internal Surfaces of an Idealized APC

From \ To	Roof	Side Wall 1	Side Wall 2	End Wall 1	End Wall 2	Floor
Roof	0	0.1625	0.1625	0.0940	0.0940	0.4800
Side Wall 1	0.3185	0	0.1600	0.1008	0.1008	0.3185
Side Wall 2	0.3185	0.1600	0	0.1008	0.1008	0.3185
End Wall 1	0.2972	0.1625	0.1625	0	0.0550	0.2972
End Wall 2	0.2972	0.1625	0.1625	0.0550	0	0.2972
Floor	0.4800	0.1625	0.1625	0.0940	0.0940	0

It should be noted that internal radiant heat transfer can be dominant over the convective heat transfer for some surfaces if the surfaces have significantly differing temperatures. This will be especially true if a warm body is present within the vehicle, such as an engine block or a soldier. If the surface temperature of the warm body is approximately the same temperature as the ambient air, convective heat transfer will be negligible. However, if the vehicle wall temperatures are significantly different from the air temperature, radiant heat transfer from the warm body will occur at an approximate rate of 1.5 W/m^2 per degree Celsius of temperature difference between the warm body and the walls. Thus, for an effective solution to the problem of maintaining a warm body inside the vehicle at a fixed temperature, the vehicle must be insulated so

If the temperature differences between the surfaces are not great, then the following approximation may be made:

$$(T_i^4 - T_j^4) = 4T_m^3(T_i - T_j) \tag{15}$$

where

$$T_m = (T_i + T_j)/2.0 \tag{16}$$

Thus equation (14) becomes:

$$Q_i = A_i \sigma T_m^3 \sum_j F_{ij}(T_i - T_j) \tag{17}$$

and an electrical analogy may be made which is similar to that in convective/conductive heat transfer. In this case, Q_i is composed of several heat transfers directly to the other walls, so the radiative thermal resistance acts in parallel to the conductive/convective thermal resistance. The radiative thermal resistance between two surfaces i and j may be defined by:

$$r_{ij} = \frac{1}{(4A_i \sigma T_m^3 F_{ij})} \tag{18}$$

Unfortunately, the problem is still non-linear due to the T_m^3 term. To eliminate the non-linearity, T_m is initially assumed to be equal to the internal air temperature. If the difference between any wall temperature and the internal air temperature is less than 5°C then the error incurred by this approximation is less than 5%. For applications in which greater accuracy is required, T_m may then be re-evaluated using the latest prediction of the wall temperatures and then iterating the solution using the improved values of the radiative thermal resistance. This procedure will also be used for external radiative heat transfer calculations.

Table 2 lists the results of the calculation of the forced convective heat transfer coefficient for the outside of the APC.

TABLE 2: Heat Transfer Coefficients for External, Turbulent Forced Convection, Heat Transfer from an Idealized APC. Free Stream Velocity, V = 20 m/s

Region	Area (m^2)	L (m)	St	h (W/m^2K)
Roof	12.0	4.2	1.31×10^{-2}	3.76×10^{2}
Side Wall	6.1	4.2	1.31×10^{-2}	3.76×10^{2}
End Wall	3.8	2.9	9.57×10^{-3}	2.75×10^{2}
Floor	12.0	4.25	1.31×10^{-2}	3.76×10^{2}

2.1.4 Radiative Heat Transfer - Internal

The amount of radiant energy leaving a black surface (ε=1) is given by [2,5]:

$$Q_i = A_i\sigma(T_i^4 - \sum_j F_{ij}T_j^4) \tag{13}$$

By conservation of energy, for any surface i the sum of the viewfactors must be one. Equation 13 therefore reduces to:

$$Q_i = A_i\sigma \sum_j F_{ij}(T_i^4 - T_j^4) \tag{14}$$

2.1.3 Convective Heat Transfer - External

In general, it cannot be expected that the air-flow over the vehicle will be small. Indeed, by the reasoning put forth in Section 2.1.1, if the air-speed over the vehicle is much larger than 1 [m/s] (3.6 [km/hr]) then the forced convection will be dominant. Improper use of a free convection analysis severely underestimates the heat transfer from the outside surfaces.

In the analysis, the following simplifying assumption is made: each surface is considered as a flat plate over which the air is blowing at the stated free-stream velocity. This assumption will over-estimate the heat transfer coefficient from the vehicle, however, as the outer boundary layers of air are found to contribute little to the total thermal resistance of the vehicle, the errors induced by this assumption are thought to be negligible.

For flow along a flat plate, the flow begins transition at a Reynolds number of approximately 1×10^5 and becomes fully turbulent when the Reynolds number reaches 2.8×10^6 [4]. If a wind speed of 20 [m/s] (72 [km/hr]) is present, nearly all of each surface (88%) will be subjected to either transitional or turbulent flow. For the purposes of this study, it will be assumed that each surface experiences turbulent flow over its entire area.

Using the Reynolds analogy, the local Stanton number can be related to the Reynolds number by [2]:

$$St_x = 0.592(Re_x)^{-\frac{1}{5}}, \qquad 5\times10^5<Re_x<1\times10^7 \tag{11a}$$

$$= 0.37(\log Re_x)^{-2.584}, \qquad 1\times10^7<Re_x<1\times10^9 \tag{11b}$$

The mean Stanton number can be determined by integrating the above relations over the length of the surface. For convenience, a wind speed of 20 [m/s] was chosen which produces a maximum Reynolds number of 8.9×10^6 which is in the first range noted above. This relationship is easily integrated and found to be:

$$St = 0.074\ L(Re_L)^{-0.2} \tag{12}$$

TABLE 1: Initial Estimates of Heat Transfer Coefficients for Internal, Free Convection, Heat Transfer in an Idealized APC

Region	Area (m^2)	L (m)	GrPr	Flow Type	Nu	h (W/m^2K)
Roof	9.8	3.225	3.09×10^{10}	Turbulent	4.39×10^2	3.29
Side Wall	5.0	1.25	8.81×10^9	Turbulent	2.07×10^2	4.00
End Wall	3.1	1.25	8.81×10^9	Turbulent	2.07×10^2	4.00
Floor	9.8	3.225	3.09×10^9	Laminar	1.13×10^2	0.85

vehicle. If it is found that larger air velocities over the interior surfaces are present, then the conclusions drawn from the following analysis may no longer be valid.

For free convection, the mean Nusselt number has been found to correlate well on a simple function of the Grashof and Prandtl numbers [2]:

$$Nu = c(GrPr)^n, \qquad 10^{-4} < GrPr < 10^{13} \tag{7}$$

which, when evaluated, gives the heat transfer coefficient:

$$h = Nu \times k \tag{8}$$

The constants "c" and "n" may be obtained from Table B-1 for free convection from vertical surfaces and from Table B-2 for free convection from horizontal surfaces, which are found in Appendix B. Table 1 gives the results of the analysis for the internal convective heat transfer coefficients.

The thermal resistance of this boundary layer of air can be calculated for each wall from the equation:

$$R = \frac{1}{hA} \tag{9}$$

2.1.2 Conductive Heat Transfer Through the Walls

The thermal resistance of a wall (R) having a known thermal conductivity (k), thickness (t) and area (A) is given by:

$$R = \frac{t}{kA}$$

It is assumed that the thermal conductivity of the aluminum [3] in the walls, floor and roof of the APC has a thermal conductivity of 204 [W/mK] and that all of these surfaces have a thickness of 0.1 [m].

A computer program has been written (Appendix C) which solves this set of simultaneous equations given the values of thermal resistance and ambient temperatures required to evaluate the elements of [A] and [B]. The program provides the total heat transfer rate for the enclosure as well as the individual heat transfer rates through each surface. Interior surface temperatures are also computed and presented (Appendix D).

The remainder of this section is devoted to the evaluation of the heat transfer coefficients and the thermal resistances for the model.

2.1.1 Convective Heat Transfer - Internal

Heat transfer by convection falls within two categories: Free Convection and Forced Convection. In free convection, the fluid velocity generated by the fluid temperature gradient is dominant over any impressed fluid velocity. In forced convection, the opposite is the case.

For most of the surfaces in the vehicle, any impressed velocity will move the air in a direction which is perpendicular to the direction caused by natural convection. Thus, the resulting heat transfer could conceivably be significantly greater than that predicted by either analysis singly. It is difficult to determine which heat transfer mode is dominant or even significant due to the lack of data on air velocities inside the vehicle. It has been observed in a tentage research study [1] that air velocities across the floor and next to the walls are typically in the range of 0 to 0.5 [m/s] for impressed circulation rates of approximately 1 [m^3/s]. A measure of the importance of each mode of convective heat transfer is given by the ratio of the Grashof number to the square of the Reynolds number. This is indicative of the importance of buoyancy forces to inertial forces. If this ratio is of the order of one, then both convection modes are important; if it is significantly greater than one, then free convection is dominant. The magnitude of the ratio is proportional to the inverse of the square of the free-stream air velocity. The value of this ratio was found to be approximately eight when an impressed air velocity of 0.5 [m/s] was assumed. This would indicate that both forms, natural and forced convective heat transfer, are approximately of equal importance and neither one should be disregarded. Unfortunately, as noted above, there is no information on internal APC air velocities on which to make valid estimates of the forced convective heat transfer. Therefore, for the purpose of this study, only the free convection will be considered with the understanding that it may underestimate the internal heat transfer. When the required data on internal air velocities are available, it is recommended that this portion of the analysis be examined to determine the error introduced by considering only the free convection and neglecting the forced convection on the internal surfaces of the

outside walls and the inner-most surfaces of any insulating cover; T_{i3} are the outside surface temperatures of any insulating cover.

Using conventional circuit analysis techniques, the problem is reduced to solving a set of simultaneous equations. The inside wall temperatures of matrix [T] are the primary unknowns which can be expressed in terms of the known thermal resistances of matrix [A] and the known internal and external temperatures of matrix [B].

At each of the 6 nodes (11,21,31,41,51,61), which correspond to six interior walls, a heat flow balance to other nodes is performed:

$$q_{oi} = q_{i\infty} + \sum_{j=1}^{6} q_{ij}$$

The heat flows are then replaced by the appropriate temperature differences and thermal resistances between nodes according to the relationship:

$$q = \frac{\Delta T}{R}$$

Upon separation of the individual wall temperatures and simplifying, a set of simultaneous equations is found:

$$[A][T] = [B] \qquad (6a)$$

where,

$$A(i,j) = -\left(\frac{1}{R_{i1}} + \frac{1}{(R_{i2} + R_{i3} + R'_{i4})} + \sum_{k} \frac{1}{r_{ik}}\right), \quad i = j \qquad (6d)$$

$$= \frac{1}{r_{ij}} \qquad , \quad i \neq j \qquad (6c)$$

$$B(i) = -\left(\frac{T_o}{R_{i1}} + \frac{T_\infty}{(R_{i2} + R_{i3} + R_{i4})}\right) \qquad (6d)$$

Having obtained the temperatures of [T], it is a simple procedure to obtain the various heat flow components. The procedure is completely analogous to determining the unknown currents in an electrical circuit analysis.

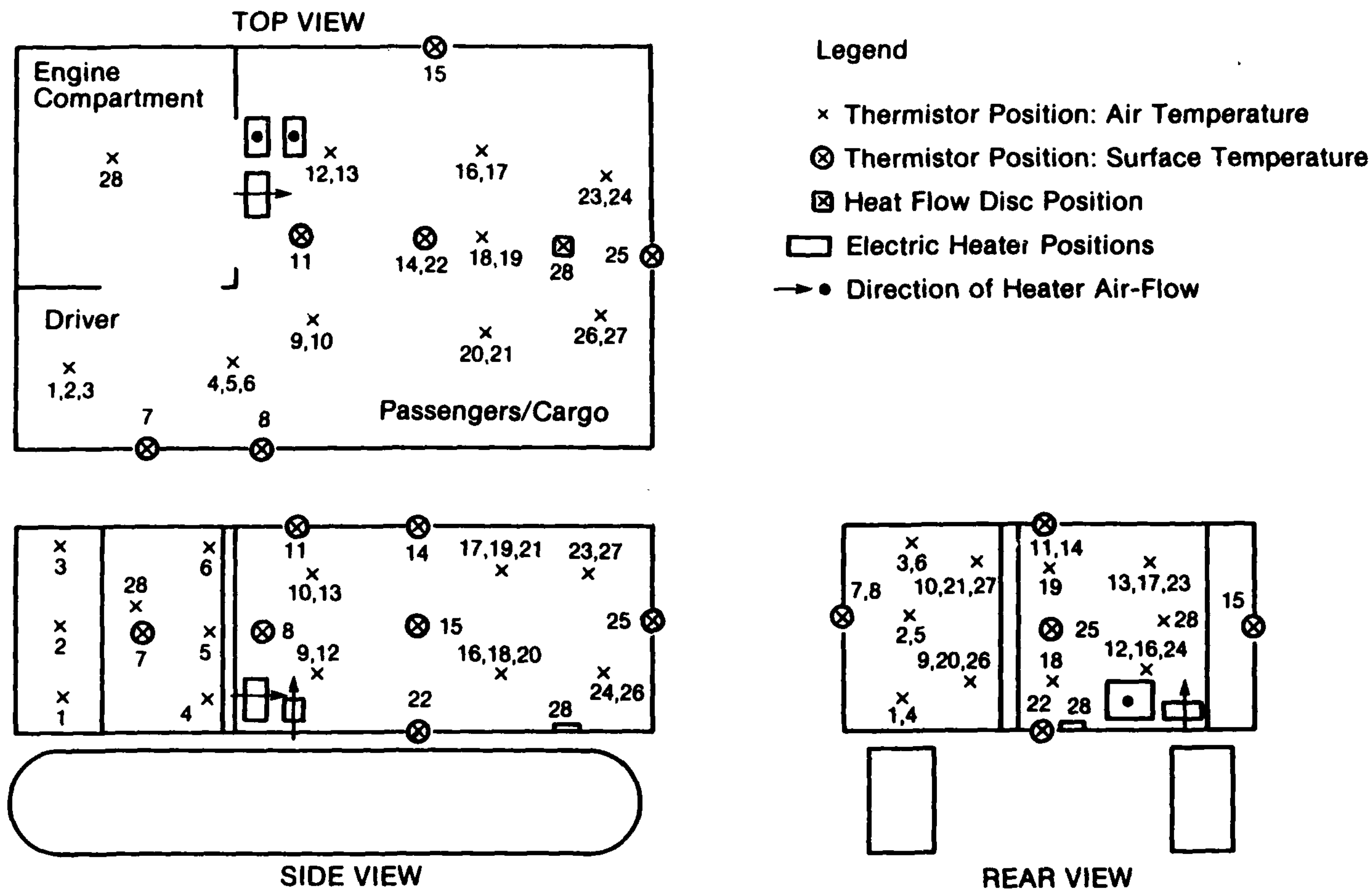

Figure 3: Typical Experimental Layout of Sensors and Equipment in the Armoured Personnel Carriers.

vehicles occurs, R_{eq} is a function of both the ventilation rate as well as the rate at which heat is lost by convection, conduction and radiation.

3.0 RESULTS

Table 5 contains the analytical estimates of the thermal energy requirements to maintain a 15°C internal temperature with an external ambient temperature of -40°C. It is assumed that no ventilation occurs. These values are taken or derived from the data listed in Appendix D. The "Test Number" indicated with the data in Appendix D is chosen such that the first digit corresponds to the case number and the second digit indicates the iteration number.

It was found that the value of the wall temperature initially assumed was often significantly different from the predicted wall temperature. Thus, the heat transfer coefficients were recalculated using improved wall temperature estimate and the analysis was repeated until the difference between two successive predictions of wall temperatures were less than 1°C. This difference results in less than a 2% error in the calculation of the heat transfer coefficients. Three iterations were usually required to achieve this precision.

TABLE 5: Predicted Values of Heating Requirement(Q), Mean Wall Temperature (T_s) and Equivalent Thermal Resistance (R) of an Idealized APC for each Insulating Case defined in Table 4c.
(T_o = 15°C, T_∞ = -40°C).

Case	Q [W]	T_s [°C]	R $[\frac{m^2K}{W}]$
1	9515	-39.3	0.21
2	4174	-20.7	0.47
3	2866	- 9.7	0.69
4	1777	- 2.4	1.11
5	1030	3.6	1.91

For each insulating case of Table 4c, a mean thermal resistance is calculated for the entire vehicle by dividing the difference between the internal and ambient air temperatures by the computed heat loss. Also,an area-weighted mean interior surface temperature (T_s) is calculated.

Examination of the results indicates that merely considering the thermal energy requirements to maintain the difference between the internal and ambient air temperatures will given an incomplete view of the heat transfer problem. The importance of the interior wall temperature can be studied with reference to the five cases defined earlier.

The analysis of Case 1 (uninsulated vehicle) indicates that it requires 9515 [W] to maintain the internal air of a temperature 55°C higher than the ambient air. However, most of the vehicle's thermal insulation results from the internal air boundary layers which form on the vehicle walls. This results in a mean internal wall surface temperature of -39.3°C which is approximately equal to the ambient temperature (-40°C). If a body which has a surface temperature which is comparable to the internal air temperature is placed into the vehicle, the convective heat transfer from that body will be negligible. However, thermal energy would be lost from the body by radiative heat transfer to the walls at an initial rate of 222 [W/m^2]. As a guide, the heat production of a typical, inactive person is approximately 45 [W/m^2]. Thus, it is apparent that the internal wall surface temperature should be approximately equal to the warm body surface temperature to minimize radiative heat ransfer from the body.

In the second case, (modeling the addition of the DRES "Blanket"), the heating requirements are reduced 44% from the uninsulated case. The highest heat loss area is now the floor, which has no additional insulation. The temperature of the floor (-39.4°C) is still very close to the ambient air temperature compared with the other interior surface temperatures (-14°C). The initial radiative heat loss from the body in the vehicle would be reduced to approximately 161 [W/m^2]. By adding an insulating covering to the floor (the equivalent of 2.2 cm of a closed cell, polyethylene foam pad) the heat loss would be reduced a further 14% from the uninsulated case, but the floor temperature would be increased significantly to -12°C and the other wall temperatures would also increase substantially to -10°C. This step would reduce the initial radiative heat loss from the body to 118 [W/m^2].

Insulating cases 4 and 5 demonstrate the effects of doubling and quadrupling the thickness of insulation of case 3. Case 5, which is the equivalent of 6 cm of the rayon insulation of the DRES "Blanket" and 8 cm of the closed cell foam pads, allows the vehicle to maintain the 55°C temperature difference with only 11% of the energy required for uninsulated vehicle. The mean wall temperature in the vehicle for this case is approximately 3.6°C, or 43.6°C above ambient. Thus, the steady-state temperature of a non-heat-producing body placed in the vehicle would be somewhere between 3.6°C and 15°C. For people, this higher wall temperature would mean less heat loss by radiation and hence a more comfortable

environment. The initial radiative heat transfer from the body would be 87 and 58 [W/m^2] for cases 4 and 5 respectively.

Table 6 gives examples of the additional heating requirements if there were to be an air exchange between the interior of the vehicle and the surroundings. The ventilation rate will be a complex function of the number and size of openings in the vehicle, the wind velocity, and the temperature difference.

TABLE 6: Predicted Heat Loss Rates for Hypothetical Air Infiltration Rates.
(T_0 = 15°C, T_∞ = -40°C)

Infiltration Rate [m^3/s]	Rate of Heat Loss [W]
0.01	837.9
0.10	8379.2
1.00	83791.6

Since no data is currently available on the ventilation rates, a typical value was assumed in order to demonstrate the significance of this form of heat loss. A minimum ventilation rate was derived by assuming that each person in the vehicle requires 1 l/s of fresh air to keep the carbon dioxide levels in the vehicle below 1%, and by assuming that there were ten people in the vehicle. Thus, the minimum ventilation rate would typically be 10 l/s or 10^{-2} [m^3/s]. From Table 6, it is seen that this requires an additional 838 W of thermal energy be supplied to maintain the same mean air temperature as when no ventilation occurred. If the hatches were open, it is expected that the ventilation rate could be much higher than the value used in this example. Again, it must be emphasized that this additional energy input would only preserve the mean air temperature and it can be expected that regions of very cold and very warm air would be present in the vehicle.

The results of the experimental investigation of the effectiveness of the DRES "Blanket" are given in Table 7. In most cases, the weather was clear with no wind. As previously noted, several different "insulating" conditions were investigated and each variation is noted in the table.

The first set of measurements indicate that there are only minor differences in the heat transfer properties of each of the uninsulated vehicles. The calculated value of the equivalent thermal resistance (R_{eq}) indicate a 3.6% difference between vehicles and it is assumed that the data analysis technique is appropriate.

The value of R_{eq} for the uninsulated vehicle (0.235[m^2K/W]) is in close agreement with that predicted by the analytical model (0.21[m^2K/W]).

The addition of the DRES "Blanket" (cases 2a and 2b) is seen to result in an increase of the R_{eq} of an uninsulated APC by approximately 24% with no wind. The values of R_{eq} (0.2908 and 0.3119 [m^2K/W] are significantly less than the predicted value (0.47 [m^2K/W]).

Air infiltration between the vehicle and the "Blanket" may be responsible for this discrepancy, however, air movement in the vehicle and ventilation may also be responsible. As previously noted, air movement in the vehicle would cause heat to be transferred to the walls at a greater rate than that predicted by a natural convective heat transfer analaysis.

Comparison of the data for heat loss through the floor (Table 7) with the predicted heat loss (Appendix D) on a per unit area per temperature difference gives varying agreement. For the uninsulated vehicles, the average heat loss through the floor was measured to be 0.96 [W/m^2K] while the predicted heat loss was 1.66 [W/m^2K]. For the vehicle insulated with the DRES "Blanket", the measured heat loss through the floor was measured to be 1.47 [W/m^2K] while the predicted heat loss was 4.17 [W/m^2K]. The discrepancy between the measured floor heat loss and the predicted floor heat loss is probably coupled to the discrepancy between the theoretical evaluation of the heat transfer coefficient and the R_{eq} for the vehicle with the DRES "Blanket". From test case 2 (Appendix D) 73% of the heat transferred to the floor is by radiation from the other walls. If the other wall temperatures are lower than expected due to an unexpectedly low value of R_{eq}, then the radiant heat transfer to the floor may be significantly less than predicted.

In Case 3, the effect of ventilation is apparent. Neither APC's have any additional insulation. The first APC has both the Crew Commander's and the Driver's hatches open, while the second APC has only the Driver's hatch open. The results in Table 7 indicate that opening the Driver's hatch causes an apparent decrease in the R_{eq} of an uninsulated vehicle of 17%. Opening both the Crew Commander's and the Driver's hatches is seen to reduce the apparent R_{eq} of an uninsulated vehicle by 37%.

In Case 3, the wind speed was negligible and thus the ventilation rate resulted from the bouyancy of the internal air only. In general, if a wind were present, it is expected that the ventilation rate could be substantially increased, greatly increasing the heat transfer from the vehicle. Also, as the insulation of the vehicle is increased, and the heat loss through the walls is reduced, the importance of ventilation rate and its associated heat loss become proportionally larger.

TABLE 7: Results Derived from Experimental Measurements of Temperature in Two APC's

Experimental Number	Experiment Description	Mean Internal Temperatures* [C]		Equivalent Thermal Resistance [m^2K/W]		% Difference**	Floor Heat Loss [W/m^2]	
		APC 1	APC 2	APC 1	APC 2		APC 1	APC 2
1	APC 1- No Insulation APC 2- No Insulation Clear, No Wind	22.5	21.3	0.2359	0.2275	5	20.5	22.2
2a	APC 1- No Insulation APC 2- DRES Blanket	22.7	27.3	0.2384	0.2908	22	21.4	44.2
2b	Clear, No Wind	22.9	29.3	0.2409	0.3119	30	16.8	38.8
3	APC 1-No Insulation 2 Hatches Open APC 2-No Insulation Driver's Hatch Open Clear, No Wind	14.0	18.4	0.1473	0.1960	37,17***	6.97	15.1

* Mean Internal Temperature: The volume weighted mean temperature relative to ambient, $Ta = \sum_i T_i v_i / V$.

** % Difference: The percentage change in the thermal resistance of APC 2 relative to APC 1 $\left|\frac{R_2 - R_1}{R_1}\right| \times 100$

*** These percentage differences are calculated based on the equivalent thermal resistance of the corresponding APC relative to the mean equivalent thermal resistance of the uninsulated APC's measured in experiments 1, 2a, 2b ($R_{eq} = 0.2352\ m^2K/W$).

4.0 CONCLUSION

The final choice of the amount of insulation to be used and how it is to be distributed on the vehicle will depend upon space limitations, available thermal energy and the desired steady-state temperature of bodies inside the vehicle. Due to the transfer of thermal energy by both conduction and radiation, it is apparent that the insulation should be distributed so as to give a uniform internal wall temperature and that this interior wall temperature should be as close to the internal air temperature as possible.

If ventilation occurs, then the value of the insulation may quickly become of secondary importance, depending upon the ventilation rate. Even at a minimum ventilation rate of 10 l/s, and with the insulation of case 5, the heating requirements for the ventilation and conductive/radiative heat losses are of the same order of magnitude, approximately 1000 W each.

Experimental measurements of the internal temperatures of two APC's, one without insulation and one with the DRES "Blanket", indicate that this covering increases the thermal resistance of an uninsulated vehicle from 0.2352 to 0.3013 [m^2K/W]. This would allow a 22% reduction in energy input from 9851 [W] to 7690 [W] while still maintaining the 55°C temperature difference. Unfortunately, this may be of little benefit if the radiative heat transfer from a body in the vehicle is large due to the low floor temperature. From the analytical study, the performance of the DRES "Blanket" can be significantly enhanced if the floor is insulated as well.

The discrepancies between R_{eq} predicted by the model and the R_{eq} found by experimentation for the insulated vehicle are significant. When data on the airflow pattern in a vehicle are available, the validity of the internal heat transfer coefficients should be determined and corrected if necessary. Some of the discrepancy may be in the manner in which the insulating covering is put on the vehicle. A poor fit may allow air to circulate between the vehicle and the covering, reducing the effectiveness of the covering as a thermal insulator. This effect would be even more significant if the vehicle were to be exposed to a wind. Portions of the

vehicle not covered by the blanket, the hatch covers for example, may be sources of high heat loss.

It should be pointed out that if the discrepancy arises from the evaluation of the internal heat transfer coefficients, the importance of the error decreases as more insulation is added to the vehicle. If the error is a result of air infiltration, the error will increase with increasing insulation and it may vary with ambient conditions. If the discrepancy is a result of a poorly fitted covering or exposed surfaces, little change in the thermal performance of a covering should be observed as the insulating value of the covering is increased.

5.0 REFERENCES

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APPENDIX A

PHYSICAL PROPERTIES OF AIR

TABLE A1: Physical Properties of Air at Standard Atmospheric Pressure. Derived from [8,9].

Temperature T [°C]	Density ρ [kg/m^3]	Kinematic Viscosity ν [m^2/s]	Specific Heat C_p [J/kg K]	Thermal Conductivity K [W/mK]	Prandtl Number Pr
-40	1.515	0.98×10^{-5}	1005.6	0.02086	0.728
-20	1.395	1.15	1005.3	0.02252	0.721
0	1.293	1.32	1005.5	0.02411	0.716
10	1.248	1.41	1005.6	0.02490	0.713
20	1.205	1.50	1005.7	0.02570	0.710
30	1.165	1.60	1005.9	0.02648	0.707
40	1.128	1.68	1006.6	0.02724	0.705

APPENDIX B

EMPIRICAL CONSTANTS FOR CONVECTIVE HEAT TRANSFER

TABLE B1: Constants c and n of Eq. (7) for Free Convection from a Vertical Plate or Cylinder at Uniform Temperature [2].

Type of Flow	Range of $Gr_L Pr$	c	n
Laminar	10^4 to 10^9	0.59	1/4
Turbulent	10^9 to 10^{10}	0.10	1/3

TABLE B2: Constants c and n of Eq. (7) for Free Convection from a Horizontal Plate at Uniform Temperature [2].

Type of Flow	Orientation of Plate*	Range of $Gr_L Pr$	c	n
Laminar	Upper surface heated or Lower surface cooled	10^5 to 2×10^7	0.54	1/4
Turbulent	Upper surface heated or Lower surface cooled	2×10^7 to 3×10^{10}	0.14	1/3
Laminar	Lower surface heated or Upper surface cooled	3×10^5 to 3×10^{10}	0.27	1/4

* Action of the fluid on the plate.

APPENDIX C

PROGRAM LISTINGS

```
C
C
      PROGRAM HTCALC
C
C
C  THIS PROGRAM IS DESIGNED TO SOLVE THE SYSTEM OF LINEAR,
C SIMULTANEOUS HEAT TRANSFER EQUATIONS BY USING A LU DECOMPOSITION,
C BACK-SUBSTITUTION AND AN IMPROVEMENT ROUTINE.  IT IS CURRENTLY
C SET UP TO HANDLE A SET OF AS MANY AS 50 LINEAR EQUATIONS
C WITH 50 UNKNOWN TEMPERATURES.  DERIVATION OF THE ELEMENTS
C OF THE MATRICIES 'A','TIN' AND 'TOUT' ARE AS GIVEN IN THE TEXT.
C THIS PROGRAM MAKES USE OF THE FOLLOWING SUBPROGAMS:
C    DCOMS - A SUBROUTINE FOR THE MATRIX REDUCTION;
C    SOLVE - A SUBROUTINE FOR THE BACK-SUBSTITUTION;
C    IMPRV - A SUBROUTINE FOR  AN ITERATIVE IMPROVEMENT SCHEME;
C    DTPROD- A FUNCTION WHICH DETERMINES THE DOT PRODUCT OF
C                TWO VECTORS.
C
C
C     PARAMETERS:
C
C
C        A      -  A 2-D ARRAY WHICH CONTAINS THE COEFFICIENTS
C                  CORRESPONDING TO THE UNKNOWNS IN THE VECTOR T.
C                  THESE COEFFICIENTS ARE COMPOSED OF THE THERMAL
C                  RESISTANCES OF THE SYSTEM.
C
C        AREA   -  A CHARACTER ARRAY WHOSE ELEMENTS ARE SURFACE
C                  NAMES WHICH CORRESPOND TO THE SURFACE DESIGNATING
C                  ELEMENT NUMBER.
C
C        B      -  A REAL VECTOR CONTAINING THE RIGHT HAND SIDE
C                  OF THE SYSTEM OF EQUATIONS.
C
C        IND    -  AN INTEGER INDICATING WHETHER OR NOT "A" IS
C                  SINGULAR. IF IND = -1 "A" IS SINGULAR AND
C                  YOU'VE GOT PROBLEMS.  IF IND = 0 THEN "A"
C                  IS NONSINGULAR AND EVERYTHING SHOULD PROCEED.
C
C        N      -  AN INTEGER INDICATING THE NUMBER OF UNKNOWN
C                  SURFACE TEMPERATURES FOR WHICH THE PROGRAM
C                  WILL SOLVE.  IT IS NECESSARY THAT N>=NDIM
C                  SINCE IF N<NDIM THE PROBLEM IS INDETERMINANT.
C
C        NDIM   -  AN INTEGER INDICATING THE NUMBER OF LINEAR
C                  EQUATIONS IN THE MATRIX 'A'.
C
C        RCON   -  A MATRIX CONTAINING THE VALUES OF THE CONDUCTIVE
C                  AND CONVECTIVE THERMAL RESISTANCES AS DEFINED
C                  IN THE TEXT.
C
C        RRAD   -  A MATRIX CONTAINING THE VALUES OF THE RADIATIVE
C                  THERMAL RESISTANCES AS DEFINED IN THE TEXT.
C
```

```
C        T      -  A REAL VECTOR WHICH WILL ULTIMATELY CONTAIN
C                  THE DESIRED TEMPERATURES (A*T=B)
C
C        TESTNO-   AN INTEGER IDENTIFYING THE TEST NUMBER
C
C        TIN    -  THE MEAN, INTERNAL AIR TEMPERATURE OF THE VEHICLE
C
C        TOUT   -  THE EXTERIOR AMBIENT AIR TEMPERATURE
C
C
C
C     DATA INPUT SCHEDULE:
C
C     RECORD 1;               TESTNO,NDIM,N
C     RECORD 2-7;             AREA(I),I=1,6
C     RECORD 8-11;            (RCON(I,J),I=1,6),J=1,4
C     RECORD 12-16;           (RRAD(I,J),I=1,5),J=I+1,6
C     RECORD 17;              TIN,TOUT
C
C
      REAL A(50,50),B(50),T(50),D(50),Z(50)
      REAL LU(50,50),R(50),RCON(50,50),RRAD(50,50),TIN,TOUT
      REAL QCON(50),QRAD(50,50),QTOT,RESIST(50)
      CHARACTER*12, AREA(6)
      INTEGER TESTNO,I,J,K,NDIM,N
      DIMENSION NPIV(50)
C
      READ(105,*) TESTNO,NDIM,N
C
      WRITE(*,5) TESTNO
  5   FORMAT(' ','HEAT TRANSFER CALCULATIONS FOR AN IDEALIZED'
     +' APC MODEL',/,' TEST NUMBER: ',I10,//)
C
      DO 10 I=1,6
         READ(105,*) AREA(I)
 10   CONTINUE
C
      READ(105,*) ((RCON(I,J),I=1,6),J=1,4)
C
      WRITE(*,15) ((AREA(I),RCON(I,3)),I=1,6)
 15   FORMAT(' ADDED INSULATION VALUES (K/W)',/,
     +6('  R(',A12,')= ',F10.6,/))
C
      DO 30 I=1,5
         READ(105,*) (RRAD(I,K),K=I+1,6)
C
         DO 20 J=I,6
C
            IF (I .EQ. J) THEN
C
C     SET RRAD(L,L)=1E99 WHICH IS INFINITY.  IT IS ACTUALLY
C     INFINITE FOR NON-CONCAVE SURFACES AS THE VIEWFACTOR FOR
C     NON-CONCAVE SURFACES IS ZERO.
C
```

```
                  RRAD(I,J) = 1.E99
C
                  ELSE
                  RRAD(J,I) = RRAD(I,J)
                  END IF
C
 20            CONTINUE
 30         CONTINUE
C
            RRAD(6,6) = 1.E99
C
            READ(105,*) TIN,TOUT
C
C            NOW COMPOSE THE ELEMENTS OF THE MATRICES 'A' AND 'B'
C
            DO 70 I=1,6
               DO 60 J=I,6
C
                  IF (I .EQ. J) THEN
C
                     RESIST(I) = 0.0
                     DO 40 K=2,4
                        RESIST(I) = RESIST(I) + RCON(I,K)
 40                  CONTINUE
                     A(I,J) = -(1./RCON(I,1) + 1./RESIST(I))
C
                     DO 50 K=1,6
                        A(I,J)=A(I,J)-1./RRAD(I,K)
 50                  CONTINUE
C
                  ELSE
                     A(I,J) = 1./RRAD(I,J)
                     A(J,I) = A(I,J)
                  END IF
C
 60            CONTINUE
 70         CONTINUE
C
            DO 90 I=1,6
               B(I) = -TIN/RCON(I,1)-TOUT/RESIST(I)
 90         CONTINUE
C
            DO 110 I=1,NDIM
               DO 100 J=1,N
                  LU(I,J)=A(I,J)
 100           CONTINUE
 110        CONTINUE
C
C          NOW PERFORM THE GAUSSIAN ELIMINATION (LU DECOMPOSITION)
C
            CALL DCOMS(NDIM,N,LU,NPIV,D,IND)
C
C           CHECK TO SEE IF THE MATRIX IS SINGULAR
C
```

```
HEAT TRANSFER CALCULATIONS FOR AN IDEALIZED APC MODEL
TEST NUMBER:          22


ADDED INSULATION VALUES (K/W)
 R(ROOF          )=     .035290
 R(SIDE WALL 1 )=     .069430
 R(SIDE WALL 2 )=     .069430
 R(END WALL 1   )=     .111450
 R(END WALL 2   )=     .111450
 R(FLOOR         )=     .000000

CONDITION NUMBER =            1.698

INTERNAL WALL SURFACE TEMPERATURES (C)

T(ROOF          )=    -12.89
T(SIDE WALL 1 )=    -14.30
T(SIDE WALL 2 )=    -14.30
T(END WALL 1   )=    -14.09
T(END WALL 2   )=    -14.09
T(FLOOR         )=    -39.41

HEAT LOSS THROUGH THE WALLS (W)

 Q(ROOF          )=   762.53
 Q(SIDE WALL 1 )=   367.46
 Q(SIDE WALL 2 )=   367.46
 Q(END WALL 1   )=   230.19
 Q(END WALL 2   )=   230.19
 Q(FLOOR         )=  2216.35

RADIANT HEAT TRANSFER BETWEEN WALLS (W)

 QRAD(ROOF           ,SIDE WALL 1 )=     12.16
 QRAD(ROOF           ,SIDE WALL 2 )=     12.16
 QRAD(ROOF           ,END WALL 1   )=      6.03
 QRAD(ROOF           ,END WALL 2   )=      6.03
 QRAD(ROOF           ,FLOOR         )=    676.65
 QRAD(SIDE WALL 1 ,SIDE WALL 2 )=       .00
 QRAD(SIDE WALL 1 ,END WALL 1   )=      -.55
 QRAD(SIDE WALL 1 ,END WALL 2   )=      -.55
 QRAD(SIDE WALL 1 ,FLOOR         )=    216.89
 QRAD(SIDE WALL 2 ,END WALL 1   )=      -.55
 QRAD(SIDE WALL 2 ,END WALL 2   )=      -.55
 QRAD(SIDE WALL 2 ,FLOOR         )=    216.89
 QRAD(END WALL 1   ,END WALL 2   )=       .00
 QRAD(END WALL 1   ,FLOOR         )=    126.53
 QRAD(END WALL 2   ,FLOOR         )=    126.53

TOTAL HEAT LOSS BY CONVECTION AND RADIATION IS:     4174.20 (W)

MEAN THERMAL INSULATION=    .013176 (K/W)
 *STOP*
```

```
HEAT TRANSFER CALCULATIONS FOR AN IDEALIZED APC MODEL
TEST NUMBER:          21


ADDED INSULATION VALUES (K/W)
 R(ROOF          )=    .035290
 R(SIDE WALL 1 )=    .069430
 R(SIDE WALL 2 )=    .069430
 R(END WALL 1   )=    .111450
 R(END WALL 2   )=    .111450
 R(FLOOR         )=    .000000

CONDITION NUMBER =              2.449

INTERNAL WALL SURFACE TEMPERATURES (C)

T(ROOF          )=   -12.26
T(SIDE WALL 1 )=   -14.19
T(SIDE WALL 2 )=   -14.19
T(END WALL 1   )=   -13.99
T(END WALL 2   )=   -13.99
T(FLOOR         )=   -39.41

HEAT LOSS THROUGH THE WALLS (W)

 Q(ROOF          )=    780.10
 Q(SIDE WALL 1 )=    368.98
 Q(SIDE WALL 2 )=    368.98
 Q(END WALL 1   )=    231.10
 Q(END WALL 2   )=    231.10
 Q(FLOOR         )=   2234.80

RADIANT HEAT TRANSFER BETWEEN WALLS (W)

 QRAD(ROOF          ,SIDE WALL 1 )=     16.64
 QRAD(ROOF          ,SIDE WALL 2 )=     16.64
 QRAD(ROOF          ,END WALL 1  )=      8.64
 QRAD(ROOF          ,END WALL 2  )=      8.64
 QRAD(ROOF          ,FLOOR       )=    692.45
 QRAD(SIDE WALL 1 ,SIDE WALL 2 )=       .00
 QRAD(SIDE WALL 1 ,END WALL 1  )=      -.54
 QRAD(SIDE WALL 1 ,END WALL 2  )=      -.54
 QRAD(SIDE WALL 1 ,FLOOR       )=    217.76
 QRAD(SIDE WALL 2 ,END WALL 1  )=      -.54
 QRAD(SIDE WALL 2 ,END WALL 2  )=      -.54
 QRAD(SIDE WALL 2 ,FLOOR       )=    217.76
 QRAD(END WALL 1   ,END WALL 2  )=       .00
 QRAD(END WALL 1   ,FLOOR       )=    127.02
 QRAD(END WALL 2   ,FLOOR       )=    127.02

TOTAL HEAT LOSS BY CONVECTION AND RADIATION IS:     4215.06 (W)

MEAN THERMAL INSULATION=     .013048 (K/W)
 *STOP*
```

```
HEAT TRANSFER CALCULATIONS FOR AN IDEALIZED APC MODEL
TEST NUMBER:          20


ADDED INSULATION VALUES (K/W)
 R(ROOF          )=    .035290
 R(SIDE WALL 1 )=    .069430
 R(SIDE WALL 2 )=    .069430
 R(END WALL 1   )=    .111450
 R(END WALL 2   )=    .111450
 R(FLOOR         )=    .000000

CONDITION NUMBER =            3.491

INTERNAL WALL SURFACE TEMPERATURES (C)

T(ROOF          )=   -18.16
T(SIDE WALL 1 )=   -14.92
T(SIDE WALL 2 )=   -14.92
T(END WALL 1   )=   -14.72
T(END WALL 2   )=   -14.72
T(FLOOR         )=   -39.56

HEAT LOSS THROUGH THE WALLS (W)

 Q(ROOF          )=   614.22
 Q(SIDE WALL 1 )=   358.59
 Q(SIDE WALL 2 )=   358.59
 Q(END WALL 1   )=   224.60
 Q(END WALL 2   )=   224.60
 Q(FLOOR         )=  1659.74

RADIANT HEAT TRANSFER BETWEEN WALLS (W)

 QRAD(ROOF         ,SIDE WALL 1 )=   -28.01
 QRAD(ROOF         ,SIDE WALL 2 )=   -28.01
 QRAD(ROOF         ,END WALL 1  )=   -17.18
 QRAD(ROOF         ,END WALL 2  )=   -17.18
 QRAD(ROOF         ,FLOOR       )=   545.89
 QRAD(SIDE WALL 1 ,SIDE WALL 2 )=      .00
 QRAD(SIDE WALL 1 ,END WALL 1  )=     -.53
 QRAD(SIDE WALL 1 ,END WALL 2  )=     -.53
 QRAD(SIDE WALL 1 ,FLOOR       )=   212.80
 QRAD(SIDE WALL 2 ,END WALL 1  )=     -.53
 QRAD(SIDE WALL 2 ,END WALL 2  )=     -.53
 QRAD(SIDE WALL 2 ,FLOOR       )=   212.80
 QRAD(END WALL 1  ,END WALL 2  )=      .00
 QRAD(END WALL 1  ,FLOOR       )=   124.12
 QRAD(END WALL 2  ,FLOOR       )=   124.12

TOTAL HEAT LOSS BY CONVECTION AND RADIATION IS:     3440.34 (W)

MEAN THERMAL INSULATION=    .015987 (K/W)
 *STOP*
```

```
HEAT TRANSFER CALCULATIONS FOR AN IDEALIZED APC MODEL
TEST NUMBER:          11


ADDED INSULATION VALUES (K/W)
 R(ROOF        )=    .000000
 R(SIDE WALL 1 )=    .000000
 R(SIDE WALL 2 )=    .000000
 R(END WALL 1  )=    .000000
 R(END WALL 2  )=    .000000
 R(FLOOR       )=    .000000

CONDITION NUMBER =           1.801

INTERNAL WALL SURFACE TEMPERATURES (C)

T(ROOF        )=   -38.90
T(SIDE WALL 1 )=   -39.28
T(SIDE WALL 2 )=   -39.28
T(END WALL 1  )=   -39.07
T(END WALL 2  )=   -39.07
T(FLOOR       )=   -39.76

HEAT LOSS THROUGH THE WALLS (W)

 Q(ROOF        )=  4148.06
 Q(SIDE WALL 1 )=  1385.06
 Q(SIDE WALL 2 )=  1385.06
 Q(END WALL 1  )=   850.51
 Q(END WALL 2  )=   850.51
 Q(FLOOR       )=   895.53

RADIANT HEAT TRANSFER BETWEEN WALLS (W)

 QRAD(ROOF        ,SIDE WALL 1 )=     3.30
 QRAD(ROOF        ,SIDE WALL 2 )=     3.30
 QRAD(ROOF        ,END WALL 1  )=      .86
 QRAD(ROOF        ,END WALL 2  )=      .86
 QRAD(ROOF        ,FLOOR       )=    21.99
 QRAD(SIDE WALL 1 ,SIDE WALL 2 )=      .00
 QRAD(SIDE WALL 1 ,END WALL 1  )=     -.57
 QRAD(SIDE WALL 1 ,END WALL 2  )=     -.57
 QRAD(SIDE WALL 1 ,FLOOR       )=     4.15
 QRAD(SIDE WALL 2 ,END WALL 1  )=     -.57
 QRAD(SIDE WALL 2 ,END WALL 2  )=     -.57
 QRAD(SIDE WALL 2 ,FLOOR       )=     4.15
 QRAD(END WALL 1  ,END WALL 2  )=      .00
 QRAD(END WALL 1  ,FLOOR       )=     3.45
 QRAD(END WALL 2  ,FLOOR       )=     3.45

TOTAL HEAT LOSS BY CONVECTION AND RADIATION IS:    9514.72 (W)

MEAN THERMAL INSULATION=    .005781 (K/W)
 *STOP*
```

```
HEAT TRANSFER CALCULATIONS FOR AN IDEALIZED APC MODEL
TEST NUMBER:         10


ADDED INSULATION VALUES (K/W)
 R(ROOF          )=    .000000
 R(SIDE WALL 1 )=    .000000
 R(SIDE WALL 2 )=    .000000
 R(END WALL 1   )=    .000000
 R(END WALL 2   )=    .000000
 R(FLOOR         )=    .000000

CONDITION NUMBER =            2.145

INTERNAL WALL SURFACE TEMPERATURES (C)

T(ROOF          )=   -39.53
T(SIDE WALL 1 )=   -39.44
T(SIDE WALL 2 )=   -39.44
T(END WALL 1   )=   -39.28
T(END WALL 2   )=   -39.28
T(FLOOR         )=   -39.88

HEAT LOSS THROUGH THE WALLS (W)

 Q(ROOF          )=  1754.75
 Q(SIDE WALL 1 )=  1085.02
 Q(SIDE WALL 2 )=  1085.02
 Q(END WALL 1   )=   664.88
 Q(END WALL 2   )=   664.88
 Q(FLOOR         )=   464.87

RADIANT HEAT TRANSFER BETWEEN WALLS (W)

 QRAD(ROOF          ,SIDE WALL 1 )=      -.84
 QRAD(ROOF          ,SIDE WALL 2 )=      -.84
 QRAD(ROOF          ,END WALL 1   )=     -1.30
 QRAD(ROOF          ,END WALL 2   )=     -1.30
 QRAD(ROOF          ,FLOOR         )=      8.72
 QRAD(SIDE WALL 1 ,SIDE WALL 2 )=       .00
 QRAD(SIDE WALL 1 ,END WALL 1   )=      -.44
 QRAD(SIDE WALL 1 ,END WALL 2   )=      -.44
 QRAD(SIDE WALL 1 ,FLOOR         )=      3.79
 QRAD(SIDE WALL 2 ,END WALL 1   )=      -.44
 QRAD(SIDE WALL 2 ,END WALL 2   )=      -.44
 QRAD(SIDE WALL 2 ,FLOOR         )=      3.79
 QRAD(END WALL 1   ,END WALL 2   )=       .00
 QRAD(END WALL 1   ,FLOOR         )=      3.01
 QRAD(END WALL 2   ,FLOOR         )=      3.01

TOTAL HEAT LOSS BY CONVECTION AND RADIATION IS:    5719.42 (W)

MEAN THERMAL INSULATION=    .009616 (K/W)
 *STOP*
```

APPENDIX D

DATA FROM THEORETICAL ANALYSIS

```
C
      CALL SOLVE(NDIM,N,LU,NPIV,R,Z)
C
C THE IMPROVED APPROXIMATION IS COMPUTED
C
      DO 30 I=1,N
         X(I)=X(I)+Z(I)
   30 CONTINUE
C
      RETURN
      END
```

```
C
C
      FUNCTION DTPROD(N,X,Y)
C
C
C  THE PURPOSE OF THIS SUBPROGRAM IS TO CALCULATE THE DOT
C PRODUCT OF TWO VECTORS OF LENGTH N.  THE DOT PRODUCT IS
C ACCUMULATED IN DOUBLE PRECISION TO ENSURE MAXIMUM ACCURACY.
C
C     CALLING SEQUENCE:    SCALAR CONSTANT = DTPROD(N,X,Y)
C
C     PARAMETERS:
C
C        N     -  AN INTEGER THAT INDICATES THE LENGTH OF THE
C                 TWO VECTORS
C
C        X,Y   -  REAL VECTORS OF LENGTH N
C
C
C
      DOUBLE PRECISION XX,YY,SUM
      DIMENSION X(50),Y(50)
      SUM = 0.D0
      DO 10 I=1,N
         XX = X(I)
         YY = Y(I)
         SUM = SUM + XX*YY
   10 CONTINUE
      DTPROD = SUM
      RETURN
      END
```

```
C                           IN THE CALLING PROGRAM
C
C        N           -      AN INTEGER CONSTANT INDICATING THE NUMBER OF
C                           UNKNOWNS IN THE SYSTEM
C
C        A           -      A REAL 2-D ARRAY OF SIZE NDIM*N HOLDING THE
C                           ORIGINAL MATRIX.  THIS ARRAY IS NOT ALTERED BY
C                           IMPRV.
C
C        LU          -      A REAL 2-D ARRAY OF SIZE NDIM*N HOLDING THE
C                           LU DECOMPOSITON OF THE ORIGINAL MATRIX.  THIS
C                           ARRAY IS NOT ALTERED BY IMPRV
C
C        NPIV        -      A INTEGER VECTOR OF SIZE N HOLDING THE PIVOT
C                           INFORMATION FROM THE ELIMINATION STEP.
C
C        B           -      A REAL VECTOR OF SIZE N HOLDING RIGHT HAND
C                           SIDE OF THE ORIGINAL SYSTEM TO BE SOLVED. THIS
C                           ARRAY IS NOT ALTERED BY IMPRV.
C
C        X           -      A REAL VECTOR OF SIZE N HOLDING THE INITAL
C                           APPROXIMATE SOLUTION. ON RETURN THIS VECTOR
C                           WILL CONTAIN THE IMPROVED APPROXIMATION
C
C        Z           -      A REAL VECTOR OF SIZE N WHICH IS DECLARED BUT
C                           NOT INITIALIZED BY THE CALLING PROGRAM.  ON
C                           RETURN Z WILL CONTAIN THE CORRECTIONS TO THE
C                           GIVEN APPROXIMATE SOLUTION
C
C        R           -      A REAL VECTOR OF SIZE N WHICH IS DECLARED BUT
C                           NOT INITIALIZED BY THE CALLING PROGRAM.  ON
C                           RETURN "R" WILL CONTAIN THE RESIDUAL R=B-AX
C                           WHERE X IS THE INITIAL APPROXIMATE SOLUTION.
C                           DOUBLE PRECISION ARITHMETIC IS USED IN
C                           COMPUTING "R"
C
C
      REAL*8 AA,XX,SUM
      DIMENSION A(50,50),B(50),X(50),Z(50),R(50),NPIV(50)
      REAL LU(50,50)
C
C CALCULATE THE RESIDUALS
C
      DO 20 I=1,N
         SUM = B(I)
         DO 10 J=1,N
            AA=A(I,J)
            XX=X(J)
            SUM=SUM-AA*XX
 10      CONTINUE
         R(I) = SUM
 20   CONTINUE
C
C THE RESIDUAL SYSTEM IS SOLVED
```

```
      X(1)=B(KPIVOT)
      DO 30 K=2,N
         KPIVOT = NPIV(K)
         KM1 = K-1
         SUM = B(KPIVOT)
         DO 20 J=1,KM1
            SUM = SUM - LU(KPIVOT,J)*X(J)
 20      CONTINUE
         X(K) = SUM
 30   CONTINUE
C
C  BACK SUBSTITUTION BEGINS
C
      X(N) = X(N)/LU(KPIVOT,N)
      K = N
      DO 50 I=2,N
         KP1 = K
         K = K-1
         KPIVOT = NPIV(K)
         SUM = X(K)
         DO 40 J = KP1,N
            SUM = SUM - LU(KPIVOT,J)*X(J)
 40      CONTINUE
         X(K) = SUM/LU(KPIVOT,K)
 50   CONTINUE
C
      RETURN
      END
```

```
C
C
      SUBROUTINE IMPRV(NDIM,N,A,LU,NPIV,B,X,Z,R)
C
C
C GIVEN AN APPROXIMATE SOLUTION FOR A LINEAR SYSTEM OF EQUATIONS
C THIS SUBROUTINE CARRIES OUT ONE ITERATION OF THE ITERATIVE
C IMPROVEMENT PROCESS FOR COMPUTING A BETTER APPROXIMATE SOLUTION.
C IT IS ASSUMED THAT BOTH THE ORIGINAL MATRIX AND ITS LU
C DECOMPOSITION ARE AVAILABLE.  THIS SUBROUTINE CAN BE USED
C IN CONJUNCTION WITH THE SUBROUTINES DCOMP, DCOMS AND SOLVE
C TO FIND THE SOLUTION OF A LINEAR SYSTEM OF EQUATIONS.
C
C     TAKEN FROM: STUNT MANUAL, DEPT OF COMPUTER SCIENCE, U OF T
C
C     CALLING SEQUENCE:    CALL IMPRV(NDIM,N,A,LU,NPIV,B,X,Z,R)
C
C     PARAMETERS:
C
C        NDIM     -  AN INTEGER CONSTANT INDICATING THE NUMBER OF
C                    ROWS IN THE ARRAYS, "A" AND "LU" AS DECLARED
```

```
C
      SUBROUTINE SOLVE(NDIM,N,LU,NPIV,B,X)
C
C
C   THIS SUBROUTINE PERFORMS THE FORWARD AND BACKWARD SUBSTITUTIONS
C STEPS IN THE SOLUTION OF A SYSTEM OF LINEAR EQUATIONS AX=B.
C IT ASSUMES THAT THE TRIANGULAR (OR LU) FACTORIZATION OF A HAS
C ALREADY BEEN COMPUTED BY, SAY, THE ROUTINE DCOMP OR DCOMS.  IF
C EITHER ROUTINE INDICATES THAT "A" IS SINGULAR, THEN THE USE OF
C SOLVE MAY PRODUCE AN OVERFLOW INDICATION.
C
C
C   REPRODUCED FROM: STUNT MANUAL, DEPT OF COMPUTER SCIENCE, U OF T
C
C
C     CALLING SEQUENCE:   CALL SOLVE(NDIM,N,LU,NPIV,B,X)
C
C     PARAMETERS:
C
C        NDIM  -  AN INTEGER INDICATING THE NUMBER OF ROWS IN THE
C                 ARRAY "A" AS DECLARED IN THE CALLING PROGRAM
C
C        N     -  AN INTEGER CONSTANT INDICATING THE SIZE OF THE
C                 SYSTEM TO BE SOLVED.
C
C        LU    -  A REAL 2-D ARRAY OF SIZE NDIM*N CONTAINING THE
C                 LU DECOMPOSITON OF A.  THIS ARRAY IS NOT ALTERED
C                 BY SOLVE.
C
C        NPIV  -  AN INTEGER VECTOR, OF DIMENSION N, HOLDING THE
C                 PIVOT INFORMATION FOR THE ELIMINATION STEP.
C
C        X     -  A REAL VECTOR, OF SIZE N, THAT IS DECLARED BUT
C                 NOT INITIALIZED BY THE CALLING PROGRAM. ON
C                 RETURN, THIS ARRAY CONTAINS THE COMPUTED
C                 SOLUTION OF THE SYSTEM.
C
C        B     -  A REAL VECTOR, OF SIZE N, HOLDING THE RIGHT HAND
C                 SIDE OF THE ORIGINAL SYSTEM TO BE SOLVED.  THE
C                 CONTENTS OF THIS VECTOR ARE UNALTERED BY SOLVE.
C
C
      DIMENSION B(50),X(50),NPIV(50)
      REAL LU(50,50)
C
C  CHECK FOR SYSTEM OF ONLY ONE UNKNOWN
C
      IF (N.GT.1) GOTO 10
      X(1) = B(1)/LU(1,1)
      RETURN
C
C  FORWARD ELIMINATION ON "B".  THE RESULT IS PLACED IN X
C
 10   KPIVOT = NPIV(1)
```

```
C
          DO 30 J=I,N
             IP = NPIV(J)
             HOLD = ABS(A(IP,I))/D(IP)
             IF (HOLD.LE.COLMAX) GOTO 30
             COLMAX = HOLD
             NROW = J
 30       CONTINUE
C
C  TEST FOR SINGULARITY.  THE MATRIX IS ASSUMED TO BE SINGULAR
C  IF COLMAX (THE ABS. VALUE OF THE SCALED PIVOT) IS EQUIVALENT
C  TO ZERO, IE, 1.0 + COLMAX = 1.0
C
C  IF THIS IS TRUE THEN THE ROUTINE PROCEEDS ON TO THE (I+1)TH
C  STAGE OF THE ELIMINATION
C
          IF ((1.0+COLMAX) .NE. 1.0) GOTO 40
          IND = -1
          GOTO 80
C
C  IF AN INTERCHANGE IS NECESSARY, ALTER THE PIVOT VECTOR "NPIV"
C
 40       IPIVOT = NPIV(NROW)
          IF (NROW .EQ. I) GOTO 50
          NPIV(NROW) = NPIV(I)
          NPIV(I) = IPIVOT
C
C  THE MULTIPLIERS FOR THE COMPUTATION OF THE REMAINING ROWS ARE
C  DETERMINED AND ELINIMATION IS PERFORMED.  THE VALUE OF EACH
C  MULTIPLIER IS STORED IN THE POSITION OF THE ELIMINATED
C  ELEMENT.
C
 50       IP1 = I+1
          DO 70 J=IP1,N
             JPIVOT = NPIV(J)
             AMULT = A(JPIVOT,I)/A(IPIVOT,I)
             A(JPIVOT,I) = AMULT
C
             DO 60 K=IP1,N
                A(JPIVOT,K)=A(JPIVOT,K)-AMULT*A(IPIVOT,K)
 60          CONTINUE
 70       CONTINUE
 80    CONTINUE
C
       TEST = 1. + ABS(A(NPIV(N),N))
       IF (TEST .EQ. 1.) THEN
       IND = -1
       END IF
C
       RETURN
       END
```

```
C
C     A          - A REAL 2-DIMENSIONAL ARRAY, OF SIZE NDIM*N
C                  HOLDING THE MATRIX TO BE DECOMPOSED.  ON
C                  RETURN, THE CONTENTS OF "A" ARE REPLACED
C                  BY THE LU FACTORIZATION.
C
C     NPIV       - AN INTEGER VECTOR, OF SIZE N, WHICH IS
C                  UNINITIALIZED AT THE TIME OF CALLING.  THIS
C                  ARRAY WILL RECORD THE REARRANGING OF THE
C
C     D          - A REAL VECTOR, OF SIZE N, THAT IS USED AS
C                  A WORKSPACE FOR THE SCALING OPERATION.  THIS
C                  ARRAY IS DECLARED BUT NOT INITIALIZED IN
C                  THE CALLING PROGRAM.
C
C     IND        - AN INTEGER INDICATING IF "A" IS SINGULAR OR NOT.
C                     IND = -1 ;  "SCALED" PIVOT > UNIT ROUND-OFF
C                         =  0 ;  "A" IS NONSINGULAR
C                  THIS PARAMETER IS NOT INITIALIZED AT THE TIME
C                  OF CALLING.
C
C
      DIMENSION A(50,50),D(50),NPIV(50)
      IND = 0
C
C   CHECH FOR A SYSTEM OF ONLY ONE UNKNOWN
C
      IF (N.EQ.1) RETURN
C
C   INITIALIZE PIVOT AND D VECTORS
C
      DO 20 I=1,N
         NPIV(I) = I
         D(I)=0.E0
C
C   THE LARGEST ABSOLUTE VALUES IN EACH ROW ARE RECORDED IN "D"
C
         DO 10 J=1,N
            D(I) = AMAX1(D(I),ABS(A(I,J)))
   10    CONTINUE
C
         IF (D(I).EQ.0.E0) D(I) = 1.E0
   20 CONTINUE
C
C   MAIN LOOP FOR GAUSS ELIMINATION
C
      NM1 = N-1
C
      DO 80 I=1,NM1
C
C   DETERMINE THE LARGEST "SCALED PIVOT  IE,
C        MAX  A(J,I)/D(J) , I>=J>=N
C
         COLMAX = 0.0
```

```
  270  CONTINUE
C
  280  FORMAT('  QRAD(',A12,',',A12,')= ',F8.2)
C
       WRITE(*,290) QTOT
  290  FORMAT(/,' TOTAL HEAT LOSS BY CONVECTION AND RADIATION '
      +'IS: ',F10.2,' (W)')
C
       WRITE(*,300) (TIN - TOUT)/QTOT
  300  FORMAT(/,' MEAN THERMAL INSULATION= ',F10.6,' (K/W)')
C
       RETURN
       END
```

```
C
C
       SUBROUTINE DCOMS(NDIM,N,A,NPIV,D,IND)
C
C
C   THIS SUBROUTINE, DECOMPOSITION WITH SCALED PARTIAL PIVOTING,
C DOES GAUSSIAN ELIMINATION OR, EQUIVALENTLY, A TRIANGULAR (LU)
C FACTORIZATION OF THE N*N MATRIX STORED IN THE ARRAY "A".  AT
C COMPLETETION, THE "A" WILL CONTAIN THE LOWER TRIANGULAR MATRIX
C OF MULTIPLIERS USED IN THE ELIMINATION AS WELL AS THE UPPER
C TRIANGULAR MATRIX "U", THE RESULT OF THE ELIMINATION.
C
C   THE MATRIX IS ASSUMED TO BE SINGULAR IF EITHER SOME ROW IS
C ZERO INITIALLY, OR, SOME "SCALED" PIVOT DURING THE ELIMINATION
C IS SMALLER THAN UNIT ROUND-OFF.  IF THE FORMER HOLDS, THE
C DECOMPOSITION DOES NOT COMMENCE.  IN THE LATTER CASE, DCOMS
C WILL COMPLETE THE DECOMPOSITION BUT THE RESULTING UPPER
C TRIANGULAR MATRIX WILL BE SINGULAR.
C
C   THIS ROUTINE CAN BE USED IN CONJUNCTION WITH THE ROUTINE
C SOLVE TO FIND THE SOLUTION TO A SYSTEM OF LINEAR EQUATIONS.
C      THE CURRENT MAXIMUM SIZE OF THE SYSTEM IS 50*50
C
C   TAKEN FROM: STUNT MANUAL; DEPT OF COMPUTER SCIENCE;U OF T
C
C
C   CALLING SEQUENCE:   CALL DCOMS(NDIM,N,A,NPIV,D,IND)
C
C   PARAMETERS:
C
C      NDIM       - AN INTEGER INDICATIONG THE NUMBER OF ROWS
C                   IN THE ARRAY "A" AS DECLARED IN THE
C                   CALLING PROGRAM.
C
C      N          - AN INTEGER CONSTANT INDICATING THE SIZE OF
C                   THE SYSTEM TO BE SOLVED
```

```
      IF (IND) 120,140,140
 120  WRITE(*,130)
 130  FORMAT(' ','MATRIX IS SINGULAR, TOO BAD! PROCESS ABORTED')
      STOP
C
 140  CALL SOLVE(NDIM,N,LU,NPIV,B,T)
      CALL IMPRV(NDIM,N,A,LU,NPIV,B,T,Z,R)
C
C    CONDA GIVES A LOWER BOUND FOR THE CONDITION NUMBER
C    OF THE MATRIX
C
      CONDA=SQRT(DTPROD(N,Z,Z)/DTPROD(N,T,T))*16.**6
      WRITE(*,150) CONDA
 150  FORMAT(' ','CONDITION NUMBER = ',F15.3)
C
      WRITE(*,160)
 160  FORMAT(/,' INTERNAL WALL SURFACE TEMPERATURES (C)',/)
C
      DO 170 I=1,N
         WRITE(*,180) AREA(I),T(I)
 170  CONTINUE
 180  FORMAT(' ','T(',A12,')= ',F8.2)
C
C    NOW CALCULATE THE HEAT TRANSFER RATES THROUGH THE WALLS
C    TO THE OUTSIDE AIR,QCON(I), AND THE TOTAL HEAT LOSS RATE, QTOT
C
      QTOT = 0.0
      DO 200 I=1,6
         QCON(I) = (T(I) - TOUT)/RESIST(I)
         QTOT = QTOT + QCON(I)
 200  CONTINUE
C
C     NOW CALCULATE THE RADIANT HEAT TRANSFER BETWEEN THE WALLS
C
      DO 220 I=1,6
         DO 210 J=I+1,6
            QRAD(I,J) = (T(I)-T(J))/RRAD(I,J)
            QRAD(J,I) = -QRAD(I,J)
 210     CONTINUE
 220  CONTINUE
C
      WRITE(*,230)
 230  FORMAT(/,' HEAT LOSS THROUGH THE WALLS (W)',/)
C
      WRITE(*,240) ((AREA(I),QCON(I)),I=1,6)
 240  FORMAT(' ',' Q(',A12,')= ',F8.2)
C
      WRITE(*,250)
 250  FORMAT(/,' RADIANT HEAT TRANSFER BETWEEN WALLS (W)',/)
C
      DO 270 I=1,6
         DO 260 J=I+1,6
            WRITE(*,280) AREA(I),AREA(J),QRAD(I,J)
 260     CONTINUE
```

```
HEAT TRANSFER CALCULATIONS FOR AN IDEALIZED APC MODEL
TEST NUMBER:          30

ADDED INSULATION VALUES (K/W)
 R(ROOF          )=    .035300
 R(SIDE WALL 1  )=    .069400
 R(SIDE WALL 2  )=    .069400
 R(END WALL 1   )=    .111500
 R(END WALL 2   )=    .111500
 R(FLOOR        )=    .052000

CONDITION NUMBER =              3.158

INTERNAL WALL SURFACE TEMPERATURES (C)

T(ROOF          )=   -11.38
T(SIDE WALL 1  )=    -9.69
T(SIDE WALL 2  )=    -9.69
T(END WALL 1   )=    -9.67
T(END WALL 2   )=    -9.67
T(FLOOR        )=   -14.99

HEAT LOSS THROUGH THE WALLS (W)

 Q(ROOF          )=   804.86
 Q(SIDE WALL 1  )=   433.54
 Q(SIDE WALL 2  )=   433.54
 Q(END WALL 1   )=   269.38
 Q(END WALL 2   )=   269.38
 Q(FLOOR        )=   478.59

RADIANT HEAT TRANSFER BETWEEN WALLS (W)

 QRAD(ROOF        ,SIDE WALL 1 )=   -14.57
 QRAD(ROOF        ,SIDE WALL 2 )=   -14.57
 QRAD(ROOF        ,END WALL 1  )=    -8.52
 QRAD(ROOF        ,END WALL 2  )=    -8.52
 QRAD(ROOF        ,FLOOR       )=    92.12
 QRAD(SIDE WALL 1 ,SIDE WALL 2 )=      .00
 QRAD(SIDE WALL 1 ,END WALL 1  )=     -.05
 QRAD(SIDE WALL 1 ,END WALL 2  )=     -.05
 QRAD(SIDE WALL 1 ,FLOOR       )=    45.75
 QRAD(SIDE WALL 2 ,END WALL 1  )=     -.05
 QRAD(SIDE WALL 2 ,END WALL 2  )=     -.05
 QRAD(SIDE WALL 2 ,FLOOR       )=    45.75
 QRAD(END WALL 1  ,END WALL 2  )=      .00
 QRAD(END WALL 1  ,FLOOR       )=    26.57
 QRAD(END WALL 2  ,FLOOR       )=    26.57

TOTAL HEAT LOSS BY CONVECTION AND RADIATION IS:    2689.29 (W)

MEAN THERMAL INSULATION=    .020451 (K/W)
 *STOP*
```

```
HEAT TRANSFER CALCULATIONS FOR AN IDEALIZED APC MODEL
TEST NUMBER:          31


ADDED INSULATION VALUES (K/W)
 R(ROOF          )=    .035300
 R(SIDE WALL 1 )=    .069400
 R(SIDE WALL 2 )=    .069400
 R(END WALL 1   )=    .111500
 R(END WALL 2   )=    .111500
 R(FLOOR         )=    .052000

CONDITION NUMBER =            4.050

INTERNAL WALL SURFACE TEMPERATURES (C)

T(ROOF          )=    -6.86
T(SIDE WALL 1 )=    -9.22
T(SIDE WALL 2 )=    -9.22
T(END WALL 1   )=    -9.22
T(END WALL 2   )=    -9.22
T(FLOOR         )=   -11.97

HEAT LOSS THROUGH THE WALLS (W)

 Q(ROOF          )=   931.87
 Q(SIDE WALL 1 )=   440.23
 Q(SIDE WALL 2 )=   440.23
 Q(END WALL 1   )=   273.36
 Q(END WALL 2   )=   273.36
 Q(FLOOR         )=   536.30

RADIANT HEAT TRANSFER BETWEEN WALLS (W)

 QRAD(ROOF           ,SIDE WALL 1 )=    20.40
 QRAD(ROOF           ,SIDE WALL 2 )=    20.40
 QRAD(ROOF           ,END WALL 1   )=    11.82
 QRAD(ROOF           ,END WALL 2   )=    11.82
 QRAD(ROOF           ,FLOOR         )=   130.41
 QRAD(SIDE WALL 1 ,SIDE WALL 2 )=      .00
 QRAD(SIDE WALL 1 ,END WALL 1   )=      .01
 QRAD(SIDE WALL 1 ,END WALL 2   )=      .01
 QRAD(SIDE WALL 1 ,FLOOR         )=    23.75
 QRAD(SIDE WALL 2 ,END WALL 1   )=      .01
 QRAD(SIDE WALL 2 ,END WALL 2   )=      .01
 QRAD(SIDE WALL 2 ,FLOOR         )=    23.75
 QRAD(END WALL 1   ,END WALL 2   )=      .00
 QRAD(END WALL 1   ,FLOOR         )=    13.73
 QRAD(END WALL 2   ,FLOOR         )=    13.73

TOTAL HEAT LOSS BY CONVECTION AND RADIATION IS:     2895.34 (W)

MEAN THERMAL INSULATION=     .018996 (K/W)
 *STOP*
```

```
HEAT TRANSFER CALCULATIONS FOR AN IDEALIZED APC MODEL
TEST NUMBER:          32


ADDED INSULATION VALUES (K/W)
 R(ROOF          )=    .035300
 R(SIDE WALL 1 )=    .069400
 R(SIDE WALL 2 )=    .069400
 R(END WALL 1  )=    .111500
 R(END WALL 2  )=    .111500
 R(FLOOR         )=    .052000

CONDITION NUMBER =            3.839

INTERNAL WALL SURFACE TEMPERATURES (C)

T(ROOF          )=    -7.39
T(SIDE WALL 1 )=    -9.38
T(SIDE WALL 2 )=    -9.38
T(END WALL 1  )=    -9.38
T(END WALL 2  )=    -9.38
T(FLOOR         )=   -12.36

HEAT LOSS THROUGH THE WALLS (W)

 Q(ROOF          )=   916.86
 Q(SIDE WALL 1 )=   437.96
 Q(SIDE WALL 2 )=   437.96
 Q(END WALL 1  )=   272.00
 Q(END WALL 2  )=   272.00
 Q(FLOOR         )=   528.92

RADIANT HEAT TRANSFER BETWEEN WALLS (W)

 QRAD(ROOF          ,SIDE WALL 1 )=     17.16
 QRAD(ROOF          ,SIDE WALL 2 )=     17.16
 QRAD(ROOF          ,END WALL 1  )=      9.91
 QRAD(ROOF          ,END WALL 2  )=      9.91
 QRAD(ROOF          ,FLOOR         )=    126.63
 QRAD(SIDE WALL 1 ,SIDE WALL 2 )=       .00
 QRAD(SIDE WALL 1 ,END WALL 1  )=      -.01
 QRAD(SIDE WALL 1 ,END WALL 2  )=      -.01
 QRAD(SIDE WALL 1 ,FLOOR         )=     25.71
 QRAD(SIDE WALL 2 ,END WALL 1  )=      -.01
 QRAD(SIDE WALL 2 ,END WALL 2  )=      -.01
 QRAD(SIDE WALL 2 ,FLOOR         )=     25.71
 QRAD(END WALL 1  ,END WALL 2  )=       .00
 QRAD(END WALL 1  ,FLOOR         )=     14.89
 QRAD(END WALL 2  ,FLOOR         )=     14.89

TOTAL HEAT LOSS BY CONVECTION AND RADIATION IS:     2865.70 (W)

MEAN THERMAL INSULATION=    .019193 (K/W)
 *STOP*
```

```
HEAT TRANSFER CALCULATIONS FOR AN IDEALIZED APC MODEL
TEST NUMBER:          40


ADDED INSULATION VALUES (K/W)
 R(ROOF          )=     .070600
 R(SIDE WALL 1 )=     .138700
 R(SIDE WALL 2 )=     .138700
 R(END WALL 1   )=     .222900
 R(END WALL 2   )=     .222900
 R(FLOOR         )=     .104000

CONDITION NUMBER =             6.376

INTERNAL WALL SURFACE TEMPERATURES (C)

T(ROOF          )=     -2.44
T(SIDE WALL 1 )=     -1.17
T(SIDE WALL 2 )=     -1.17
T(END WALL 1   )=     -1.15
T(END WALL 2   )=     -1.15
T(FLOOR         )=     -5.06

HEAT LOSS THROUGH THE WALLS (W)

 Q(ROOF          )=    530.09
 Q(SIDE WALL 1 )=    278.92
 Q(SIDE WALL 2 )=    278.92
 Q(END WALL 1   )=    173.43
 Q(END WALL 2   )=    173.43
 Q(FLOOR         )=    335.09

RADIANT HEAT TRANSFER BETWEEN WALLS (W)

 QRAD(ROOF         ,SIDE WALL 1 )=   -10.93
 QRAD(ROOF         ,SIDE WALL 2 )=   -10.93
 QRAD(ROOF         ,END WALL 1   )=    -6.40
 QRAD(ROOF         ,END WALL 2   )=    -6.40
 QRAD(ROOF         ,FLOOR         )=    67.01
 QRAD(SIDE WALL 1 ,SIDE WALL 2 )=      .00
 QRAD(SIDE WALL 1 ,END WALL 1   )=     -.04
 QRAD(SIDE WALL 1 ,END WALL 2   )=     -.04
 QRAD(SIDE WALL 1 ,FLOOR         )=    33.62
 QRAD(SIDE WALL 2 ,END WALL 1   )=     -.04
 QRAD(SIDE WALL 2 ,END WALL 2   )=     -.04
 QRAD(SIDE WALL 2 ,FLOOR         )=    33.62
 QRAD(END WALL 1   ,END WALL 2   )=      .00
 QRAD(END WALL 1   ,FLOOR         )=    19.53
 QRAD(END WALL 2   ,FLOOR         )=    19.53

TOTAL HEAT LOSS BY CONVECTION AND RADIATION IS:     1769.87 (W)

MEAN THERMAL INSULATION=    .031076 (K/W)
 *STOP*
```

HEAT TRANSFER CALCULATIONS FOR AN IDEALIZED APC MODEL
TEST NUMBER: 41

ADDED INSULATION VALUES (K/W)

```
 R(ROOF        )=    .070600
 R(SIDE WALL 1 )=    .138700
 R(SIDE WALL 2 )=    .138700
 R(END WALL 1  )=    .222900
 R(END WALL 2  )=    .222900
 R(FLOOR       )=    .104000
```

CONDITION NUMBER =   3.097

INTERNAL WALL SURFACE TEMPERATURES (C)

```
T(ROOF        )=      .73
T(SIDE WALL 1 )=    -1.84
T(SIDE WALL 2 )=    -1.84
T(END WALL 1  )=    -1.86
T(END WALL 2  )=    -1.86
T(FLOOR       )=    -3.41
```

HEAT LOSS THROUGH THE WALLS (W)

```
 Q(ROOF        )=   574.81
 Q(SIDE WALL 1 )=   274.11
 Q(SIDE WALL 2 )=   274.11
 Q(END WALL 1  )=   170.26
 Q(END WALL 2  )=   170.26
 Q(FLOOR       )=   350.95
```

RADIANT HEAT TRANSFER BETWEEN WALLS (W)

```
 QRAD(ROOF        ,SIDE WALL 1 )=     22.21
 QRAD(ROOF        ,SIDE WALL 2 )=     22.21
 QRAD(ROOF        ,END WALL 1  )=     12.98
 QRAD(ROOF        ,END WALL 2  )=     12.98
 QRAD(ROOF        ,FLOOR       )=    105.66
 QRAD(SIDE WALL 1 ,SIDE WALL 2 )=       .00
 QRAD(SIDE WALL 1 ,END WALL 1  )=       .07
 QRAD(SIDE WALL 1 ,END WALL 2  )=       .07
 QRAD(SIDE WALL 1 ,FLOOR       )=     13.56
 QRAD(SIDE WALL 2 ,END WALL 1  )=       .07
 QRAD(SIDE WALL 2 ,END WALL 2  )=       .07
 QRAD(SIDE WALL 2 ,FLOOR       )=     13.56
 QRAD(END WALL 1  ,END WALL 2  )=       .00
 QRAD(END WALL 1  ,FLOOR       )=      7.72
 QRAD(END WALL 2  ,FLOOR       )=      7.72
```

TOTAL HEAT LOSS BY CONVECTION AND RADIATION IS: 1814.49 (W)

MEAN THERMAL INSULATION=   .030312 (K/W)
 STOP

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*
HEAT TRANSFER CALCULATIONS FOR AN IDEALIZED APC MODEL
TEST NUMBER:          42


ADDED INSULATION VALUES (K/W)
 R(ROOF          )=    .070600
 R(SIDE WALL 1 )=    .138700
 R(SIDE WALL 2 )=    .138700
 R(END WALL 1   )=    .222900
 R(END WALL 2   )=    .222900
 R(FLOOR         )=    .104000

CONDITION NUMBER =             3.135

INTERNAL WALL SURFACE TEMPERATURES (C)

T(ROOF          )=     -.88
T(SIDE WALL 1 )=    -2.22
T(SIDE WALL 2 )=    -2.22
T(END WALL 1   )=    -2.22
T(END WALL 2   )=    -2.22
T(FLOOR         )=    -4.07

HEAT LOSS THROUGH THE WALLS (W)

 Q(ROOF          )=   552.05
 Q(SIDE WALL 1 )=   271.40
 Q(SIDE WALL 2 )=   271.40
 Q(END WALL 1   )=   168.66
 Q(END WALL 2   )=   168.66
 Q(FLOOR         )=   344.59

RADIANT HEAT TRANSFER BETWEEN WALLS (W)

 QRAD(ROOF          ,SIDE WALL 1 )=    11.55
 QRAD(ROOF          ,SIDE WALL 2 )=    11.55
 QRAD(ROOF          ,END WALL 1   )=     6.71
 QRAD(ROOF          ,END WALL 2   )=     6.71
 QRAD(ROOF          ,FLOOR         )=    81.44
 QRAD(SIDE WALL 1 ,SIDE WALL 2 )=      .00
 QRAD(SIDE WALL 1 ,END WALL 1   )=      .01
 QRAD(SIDE WALL 1 ,END WALL 2   )=      .01
 QRAD(SIDE WALL 1 ,FLOOR         )=    16.02
 QRAD(SIDE WALL 2 ,END WALL 1   )=      .01
 QRAD(SIDE WALL 2 ,END WALL 2   )=      .01
 QRAD(SIDE WALL 2 ,FLOOR         )=    16.02
 QRAD(END WALL 1   ,END WALL 2   )=      .00
 QRAD(END WALL 1   ,FLOOR         )=     9.25
 QRAD(END WALL 2   ,FLOOR         )=     9.25

TOTAL HEAT LOSS BY CONVECTION AND RADIATION IS:    1776.77 (W)

MEAN THERMAL INSULATION=    .030955 (K/W)
 *STOP*
```

```
HEAT TRANSFER CALCULATIONS FOR AN IDEALIZED APC MODEL
TEST NUMBER:         50


ADDED INSULATION VALUES (K/W)
 R(ROOF          )=    .141200
 R(SIDE WALL 1 )=    .277400
 R(SIDE WALL 2 )=    .277400
 R(END WALL 1   )=    .445800
 R(END WALL 2   )=    .445800
 R(FLOOR         )=    .207900

CONDITION NUMBER =           1.220

INTERNAL WALL SURFACE TEMPERATURES (C)

T(ROOF          )=     4.61
T(SIDE WALL 1 )=     5.42
T(SIDE WALL 2 )=     5.42
T(END WALL 1   )=     5.43
T(END WALL 2   )=     5.43
T(FLOOR         )=     2.96

HEAT LOSS THROUGH THE WALLS (W)

 Q(ROOF          )=   315.33
 Q(SIDE WALL 1 )=   163.43
 Q(SIDE WALL 2 )=   163.43
 Q(END WALL 1   )=   101.66
 Q(END WALL 2   )=   101.66
 Q(FLOOR         )=   206.37

RADIANT HEAT TRANSFER BETWEEN WALLS (W)

 QRAD(ROOF          ,SIDE WALL 1 )=    -7.00
 QRAD(ROOF          ,SIDE WALL 2 )=    -7.00
 QRAD(ROOF          ,END WALL 1   )=    -4.10
 QRAD(ROOF          ,END WALL 2   )=    -4.10
 QRAD(ROOF          ,FLOOR         )=    42.08
 QRAD(SIDE WALL 1 ,SIDE WALL 2 )=      .00
 QRAD(SIDE WALL 1 ,END WALL 1   )=     -.03
 QRAD(SIDE WALL 1 ,END WALL 2   )=     -.03
 QRAD(SIDE WALL 1 ,FLOOR         )=    21.25
 QRAD(SIDE WALL 2 ,END WALL 1   )=     -.03
 QRAD(SIDE WALL 2 ,END WALL 2   )=     -.03
 QRAD(SIDE WALL 2 ,FLOOR         )=    21.25
 QRAD(END WALL 1   ,END WALL 2   )=      .00
 QRAD(END WALL 1   ,FLOOR         )=    12.35
 QRAD(END WALL 2   ,FLOOR         )=    12.35

TOTAL HEAT LOSS BY CONVECTION AND RADIATION IS:    1051.87 (W)

MEAN THERMAL INSULATION=    .052288 (K/W)
 *STOP*
```

```
HEAT TRANSFER CALCULATIONS FOR AN IDEALIZED APC MODEL
TEST NUMBER:          51


ADDED INSULATION VALUES (K/W)
 R(ROOF          )=    .141200
 R(SIDE WALL 1 )=    .277400
 R(SIDE WALL 2 )=    .277400
 R(END WALL 1   )=    .445800
 R(END WALL 2   )=    .445800
 R(FLOOR        )=    .207900

CONDITION NUMBER =            2.449

INTERNAL WALL SURFACE TEMPERATURES (C)

T(ROOF          )=     4.30
T(SIDE WALL 1 )=     3.19
T(SIDE WALL 2 )=     3.19
T(END WALL 1   )=     3.17
T(END WALL 2   )=     3.17
T(FLOOR        )=     2.22

HEAT LOSS THROUGH THE WALLS (W)

 Q(ROOF          )=   313.15
 Q(SIDE WALL 1 )=   155.40
 Q(SIDE WALL 2 )=   155.40
 Q(END WALL 1   )=    96.61
 Q(END WALL 2   )=    96.61
 Q(FLOOR        )=   202.84

RADIANT HEAT TRANSFER BETWEEN WALLS (W)

 QRAD(ROOF          ,SIDE WALL 1 )=     9.60
 QRAD(ROOF          ,SIDE WALL 2 )=     9.60
 QRAD(ROOF          ,END WALL 1   )=     5.62
 QRAD(ROOF          ,END WALL 2   )=     5.62
 QRAD(ROOF          ,FLOOR        )=    52.98
 QRAD(SIDE WALL 1 ,SIDE WALL 2 )=      .00
 QRAD(SIDE WALL 1 ,END WALL 1   )=      .04
 QRAD(SIDE WALL 1 ,END WALL 2   )=      .04
 QRAD(SIDE WALL 1 ,FLOOR        )=     8.33
 QRAD(SIDE WALL 2 ,END WALL 1   )=      .04
 QRAD(SIDE WALL 2 ,END WALL 2   )=      .04
 QRAD(SIDE WALL 2 ,FLOOR        )=     8.33
 QRAD(END WALL 1   ,END WALL 2   )=      .00
 QRAD(END WALL 1   ,FLOOR        )=     4.76
 QRAD(END WALL 2   ,FLOOR        )=     4.76

TOTAL HEAT LOSS BY CONVECTION AND RADIATION IS:     1020.01 (W)

MEAN THERMAL INSULATION=    .053921 (K/W)
 *STOP*
```

```
HEAT TRANSFER CALCULATIONS FOR AN IDEALIZED APC MODEL
TEST NUMBER:          52


ADDED INSULATION VALUES (K/W)
 R(ROOF          )=    .141200
 R(SIDE WALL 1 )=    .277400
 R(SIDE WALL 2 )=    .277400
 R(END WALL 1   )=    .445800
 R(END WALL 2   )=    .445800
 R(FLOOR         )=    .207900

CONDITION NUMBER =            2.449

INTERNAL WALL SURFACE TEMPERATURES (C)

T(ROOF          )=     4.55
T(SIDE WALL 1 )=     3.76
T(SIDE WALL 2 )=     3.76
T(END WALL 1   )=     3.76
T(END WALL 2   )=     3.76
T(FLOOR         )=     2.55

HEAT LOSS THROUGH THE WALLS (W)

 Q(ROOF          )=   314.91
 Q(SIDE WALL 1 )=   157.46
 Q(SIDE WALL 2 )=   157.46
 Q(END WALL 1   )=    97.91
 Q(END WALL 2   )=    97.91
 Q(FLOOR         )=   204.41

RADIANT HEAT TRANSFER BETWEEN WALLS (W)

 QRAD(ROOF          ,SIDE WALL 1 )=      6.80
 QRAD(ROOF          ,SIDE WALL 2 )=      6.80
 QRAD(ROOF          ,END WALL 1   )=      3.96
 QRAD(ROOF          ,END WALL 2   )=      3.96
 QRAD(ROOF          ,FLOOR         )=     50.96
 QRAD(SIDE WALL 1 ,SIDE WALL 2 )=       .00
 QRAD(SIDE WALL 1 ,END WALL 1   )=       .01
 QRAD(SIDE WALL 1 ,END WALL 2   )=       .01
 QRAD(SIDE WALL 1 ,FLOOR         )=     10.44
 QRAD(SIDE WALL 2 ,END WALL 1   )=       .01
 QRAD(SIDE WALL 2 ,END WALL 2   )=       .01
 QRAD(SIDE WALL 2 ,FLOOR         )=     10.44
 QRAD(END WALL 1   ,END WALL 2   )=       .00
 QRAD(END WALL 1   ,FLOOR         )=      6.02
 QRAD(END WALL 2   ,FLOOR         )=      6.02

TOTAL HEAT LOSS BY CONVECTION AND RADIATION IS:    1030.05 (W)

MEAN THERMAL INSULATION=    .053395 (K/W)
 *STOP*
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13. ABSTRACT

This report describes the heating properties of the Canadian Forces Armoured Personnel Carrier (APC). In the theoretical analysis, the APC is modelled as a box, having various insulating covers on the surfaces. The rate of heat transfer from the APC is then estimated for each insulation configuration using empirical heat transfer coefficients and measured thermal conductances. All assumptions, equations and physical data used in the analysis are presented as is a listing of the FORTRAN computer program which was used to solve the heat transfer equations. Thus, the reader may use this report as a guide to performing the heat transfer analyses for similar problems. Field measurements of internal air temperature and of heat flow-rates were made using two APC's in various insulating configurations. The measured heat loss from an uninsulated APC was in close agreement to the heat loss predicted by the theoretical analysis. However, the measured heat loss from an APC insulated with a covering developed by the Defence Research Establishment Suffield was found to be significantly greater than that predicted by the theoretical analysis. Theories are advanced on possible sources of error and discrepancy between the model predictions and the measured values of temperature and heat flow. Heat loss due to ventilation is also examined briefly.

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